MARINE STEAM TURBINES

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(*FORMING THE SUPPLEMENTARY VOLUME TO "MARINE ENGINES AND BOILERS"*)

BY

DR G. BAUER

Director of the Vulcan Works, Stettin

AND

O. LASCHE

Director of the A.E.G. Turbine Works, Berlin

ASSISTED BY

E. LUDWIG AND H. VOGEL

TRANSLATED FROM THE GERMAN AND EDITED

BY

M. G. S. SWALLOW

With 103 Illustrations and 18 Tables, including an Entropy Chart

NEW YORK

THE NORMAN W. HENLEY PUBLISHING CO.

132 NASSAU STREET

LONDON

CROSBY LOCKWOOD AND SON

1911

AUTHORS' PREFACE TO GERMAN EDITION.

IN the Preface to the Third Edition of Dr Bauer's book on the Design and Construction of Marine Engines and Boilers,* the remark is made that a book on Marine Engines is not complete without dealing with the Steam Turbine, and the publication of a supplementary volume to the third edition was promised.

Lately the use of the Steam Turbine for the propulsion of naval vessels has increased to such an extent that it appeared advisable to publish this supplementary volume sooner than originally intended.

After the A.E.G. Curtis turbine had been developed by the Allgemeine Electricitäts Gesellschaft the Stettiner Maschinenbau A.G. "Vulkan" commenced to build the same type of turbine for marine propulsion. It was the friendly relations thus established that led to the joint work of authorship of this volume, with the assistance of Messrs Wagner, Ludwig and Vogel.

As it was intended that the book should be a supplementary volume, the contents were made as concise as possible, but the authors hope to be able to extend the scope of the volume should a further edition be required. Apart from the short time available for publication, the contents could not be increased for purely business reasons, and it was further our object only to refer to types of turbines which have been proved and definitely adopted.

As far as the theoretical part of the work is concerned the contents are based on the deduction and knowledge which is

* Dr G. Bauer and Leslie S. Robertson, "Marine Engines and Boilers," translated from the second German edition, published by Crosby Lockwood & Son, London, 1905.

now the common property of scientific engineering, but for a detailed study of the theory of the steam turbine we would refer to Dr A. Stodola's book on the steam turbine.*

It was not the intention of the authors to write a similar volume for the marine engineer, and it was only the desire to promote an understanding of the calculation of steam turbines which led us to give a short description of the theory. We trust that this short description and the large number of illustrations will serve marine engineers as an introduction to the theory of marine steam turbine design.

Regarding the manner in which we hope this small volume will be judged we beg to quote again the words in the Preface to the First Edition of our book on Marine Engines referred to above:—

"In judging the book it must be remembered that it is the fruit of the few hours of available leisure snatched from the midst of a strenuous professional life. Hence it will often be found to lack the necessary finish, and to bear evidence of various shortcomings. It is, however, hoped that, as the book has been written in response to an actual daily felt need, it may for that very reason be of greater service.

"In conclusion the authors would be glad if their readers would bring to their notice any errors and omissions so that they may be rectified should, in the future, a new edition be called for."

Dr BAUER.
O. LASCHE.

STETTIN AND BERLIN, *May* 1909.

* Dr A. Stodola, "Die Dampf Turbinen," fourth German edition, 1910, published by J. Springer, Berlin.

Dr A. Stodola, "Steam Turbines," English translation of second German edition, by Dr Louis C. Loewenstein, published by Archibald Constable & Co., Ltd., London, 1905. D. Van Nostrand Co., New York.

TRANSLATOR'S PREFACE.

IN view of the reception given to Mr Robertson's translation of Dr Bauer's volume on The Marine Engine,* no apology need be offered in publishing a translation of the supplementary volume in German, by Dr Bauer and O. Lasche, on the "Marine Steam Turbine," as it is hoped that this will complete a work which has been of considerable assistance to marine engineers.

It has been the endeavour to enhance the usefulness of the volume as much as possible for English engineers, by giving all formulæ which are expressed in the German edition in Metric Units, in both Metric and English Units, so that they may be applied to problems by English engineers.

By the kind permission of Messrs Lionel S. Marks and Harvey N. Davis it has been possible for the translator to utilise the results of their labours, and embody in the translation part of the diagrams and tables published in their book on the Properties of Steam.†

This refers specially to the diagrams Figs. 3, 4, 5, 7 and Table 17 as well as the "Mollier" diagram (Entropy Chart) at the end of the book, and I wish to express my thanks to the authors for kindly granting me their permission for the use of these.

It will be noticed that in some cases there is a slight difference in the relative heat values expressed in Calories and British Thermal Units. At the time the book was written the authors had at their disposal the original Mollier diagrams

* Dr G. Bauer and Leslie S. Robertson, "Marine Engines and Boilers," published by Crosby Lockwood & Son, London, 1905.

† Lionel S. Marks and Harvey N. Davis, "Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam," published by Longmans, Green & Co., London, 1909.

published in 1906, whilst the new Entropy Chart of Messrs Marks and Davis is based on the latest researches of Knoblauch, of Thomas, and of Henning, on the values of the specific heat of superheated steam, and varies slightly from the original Mollier diagram.

The Table 17, as well as the index and table of notations, are new.

M. G. S. SWALLOW.

WEST HARTLEPOOL,
June 1911.

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NOTATIONS USED.

A - - Mechanical equivalent of heat $\frac{1}{427}$ (Metric units), $\frac{1}{778}$ (English units).
a - - Depth of blade passage.
c_p - - Specific heat of superheated steam.
c_0, c_1, c_2, Steam velocities.
D - - Mean blade diameter.
E - - Modulus of elasticity.
F - - Cross section.
g - - Acceleration due to gravity 9·81 m. per sec. or 32·2 ft. per sec.
H - - Total heat of saturated steam.
H_1 - - Total heat of superheated steam.
h - - Heat of liquid, also heat "drop."
h_1 - - Heat to be added for superheating.
I - - Moment of inertia.
K - - Heat transmission coefficient.
L - - Latent heat.
l - - Length of blades.
N - - Power in H.P. or Ch. à V.
n - - Number of revolutions or number of stages.
P - - Pressure in kg. per sq. m. or lbs. per sq. ft.
p - - Pressure in kg. per sq. cm. or lbs. per sq. in.
Q - - Quantity of steam per hour.
Q_s - - Quantity of steam per sec.
q - - Quantity of steam of auxiliaries per hour.
R - - Resistance of hull of vessel.
r - - Radius.
S - - Cooling surface.
s - - Slip, or length of path.
T - - Absolute temperature.
t - - Temperature, also pitch of blades.
u - - Blade tip velocity of propeller and blade velocity at mean blade diameter of turbine.
v - - Volume or velocities.
W - - Mechanical work in kg., m., or ft. lbs.
w - - Velocities.
x - - Dryness factor.
z - - Number of blades.
α - - Nozzle or blade angle.
δ - - Blade thickness.
η - - Efficiency.
ρ - - Friction coefficient.
ϕ - - Coefficient of velocity.
ω - - Angular velocity.

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MARINE STEAM TURBINES

PART I.

INTRODUCTION.

SECTION I.

§ 1. Advantages of Steam Turbine Installation as compared with Reciprocating Engine.

1. **Steam Consumption.**—The steam consumption of marine steam turbines running at full load is, under the most advantageous conditions, lower than that of reciprocating engines. This advantage is more pronounced, the larger the output and the higher the speed of the turbine, and is consequently more apparent in torpedo boats and small cruisers than in vessels of the mercantile marine.

2. **Weight.**—The weight of steam turbines where large outputs and high speeds are required is lower than the weight of reciprocating engines designed for the same steam consumption.

Comparing the weight of the equipment of turbine machinery with the weight of reciprocating engines in vessels of the same type, and including the complete plant consisting of boilers, plant, and propelling machinery, the relative figures are—

Torpedo boats, turbine machinery, 10 to 15 per cent. lighter.
High speed cruisers, turbine machinery, 5 to 10 per cent. lighter.
Battleships, turbine machinery equal or greater than reciprocating engines.

In the case of vessels in the mercantile marine it will only be possible in exceptionally favourable cases to keep the weight of the turbine equipment as low as that of reciprocating engines.

3. **Space.**—The space occupied by single shaft turbines (see Sec. III.) is, as a rule, less than that of reciprocating engines, but when Parsons' turbines are used it is occasionally somewhat greater than with reciprocating engines.

With all types of turbine installations the height is less than with reciprocating engines. Even taking into consideration that sufficient space must be provided for the lifting gear for dismantling the exhaust pipe connections, a less height is required than with reciprocating engines, which constitute a considerable advantage in battleships with armoured decks.

4. **Vibrations.**—All propellers cause more or less vibrations of the stern of a boat due to the great difference in the hydrostatic head of the water in which the propeller is working and to the acute angle at which the stream lines of water meet the axis of the propeller. These vibrations, which also exist in turbine vessels, make the stay in the space near the propeller most unpleasant. But there are no vibrations of high amplitude due to unbalanced rotating masses, which in the case of an imperfectly balanced reciprocating engine may cause vibrations of large amplitude of the hull of the vessel. It is only the after end of a turbine steamer which is subject to vibrations of a high periodicity and small amplitude, and which affect the midship and fore part of the vessel so little that in these parts it is difficult to feel whether the ship is in motion or not. This is a considerable advantage, especially in war vessels, as the accuracy of gun laying amidships and in the bows is greatly increased.

5. **Manœuvring.**—The ease with which steam turbines, and especially those of the single shaft system (see Sec. III.), can be manœuvred is a great advantage. For the Curtis and the A.E.G. turbine the manœuvring valve is a plain stop valve for the ahead turbine and a similar valve for the astern turbine, the handling of which is a simpler matter than that of the reversing gear for reciprocating engines. Even by unskilled hands the engines can be safely operated, a fact of particular value in war. The use of a reversing turbine cannot be dispensed with, but as the turbine equipment is superior to the reciprocating engine, as far as weight and steam consumption are concerned, the necessity for a reversing turbine need not be considered a serious disadvantage when comparing the two types of engines.

6. **Attendance.**—Quite apart from the ease with which turbine vessels can be manœuvred, the machinery is easier attended to than the reciprocating engine. Practically all those parts of reciprocating engines

which, like the bearings of the reciprocating parts, stuffing boxes, are liable to give rise to trouble and require continuous supervision are absent.

7. **Cleanliness of Operation.**—As a well-designed turbine has no steam, water, or oil leakages, the cleanness of the engine-room is marked as compared to an engine-room with reciprocating engines, dripping with oil and water in vessels under forced conditions of operation. This cleanliness is being further increased by the adoption of turbine-driven engine-room auxiliaries.

8. **Maintenance and Repairs.**—The absence of all rubbing parts such as glands, &c., entail less repairs, and a well-designed and well-supervised turbine need not be opened out for inspection except at very infrequent intervals.

SECTION II.

§ 2. THE USE OF MARINE STEAM TURBINES.

In view of the large number of requirements a marine steam turbine has to fulfil, the construction always demands compromises in the same manner as the construction of a reciprocating engine for a similar purpose. At the present state of the art the use of turbines as propelling engines for certain types of vessels is still practically impossible.

The useful limit of the marine steam turbine is roughly as follows:—

1. **Torpedo Boats.**—For this type of boat at the speeds at present demanded the turbine must certainly receive preference. The high speed of the boat allows of a high speed of revolution of the engine, whilst the large output ensures a moderate dead weight and a low steam consumption.

2. **Fast Cruisers.**—For the same reasons as for torpedo boats the turbine will be given preference for this type of vessel.

3. **Large Cruisers.**—With regard to weight and steam consumption the advantage lies with the steam turbine at high speeds, but at lower speeds, if the turbine does not prove superior under those two heads, the other advantages enumerated above (§ 1) will in most cases assist in settling the question in favour of the turbine.

4. **Battleships.**—Here the reciprocating engine is probably in most cases superior to the turbine as regards weight, and although at full speed the steam consumption of the turbine will be less than that of the reciprocating engines, at cruising speed the advantage is on the side of the latter type of engine. The general advantages (Sec. I.), as the constructive developments of the turbine progresses, will tend to make it of greater advantage to employ this engine for battleships in the future.

5. **Mercantile Marine.**—In the mercantile marine at the present the use of steam turbines is only justified for speeds above 20 knots. The reason for this is that in most cases, and except in small vessels, a comparatively low number of revolutions must be chosen to avoid too low a propeller efficiency. A low number of revolutions entails a large dead weight, a large amount of space as well as a high price for the turbine, facts which generally prohibit the use of turbines. Naturally the question of price is here of much more importance than in the case of battleships where only the best type of machinery is used. Generally speaking the first cost of a turbine equipment including boiler plant and engines is higher than that of reciprocating engines, a fact which increases the difficulty of using the new type of engine in vessels which are of the profit-earning class.

SECTION III.

§ 3. General Classification of Marine Steam Turbines.

Marine steam turbines can be divided into two classes:—

1. Turbines in which the full expansion of steam does not take place in a turbine driving a *single* shaft, but where the steam expanding from the inlet to the exhaust pressure passes through *a number* of separate turbines arranged for driving *two* or *more shafts*. All turbine installations with pure Parsons turbines are arranged on these lines, as it is not possible to build a turbine of this type of sufficient length to enable the relatively low speed to be obtained for driving a single propeller shaft. This ingenious solution of the problem of using Parsons turbines has, besides the advantage of the best possible economy, some disadvantages to which we shall refer later.

2. Turbines in which the complete expansion of steam from inlet to exhaust pressure occurs in a turbine on one shaft; so-called *single shaft turbines*. To this class of turbine belong the Curtis turbine and the turbines of the Allgemeine Electricitäts Gesellshaft referred to as the A.E.G. turbine.

Apart from these two systems no other type of marine turbine system has been practically tried on a large scale. The single shaft systems of Rateau, Zölly, Schichau, and Melms-Pfenniger are still in the experimental stage, and up to the present have not demonstrated their practical success at sea on a large scale.

PART II.

GENERAL REMARKS ON THE DESIGN OF A TURBINE INSTALLATION.

SECTION I.

Indicated and Effective Output.

§ 4. **General Remarks.**—The work exerted by the steam on the piston, that is, the part of the reciprocating engine coming in direct contact with the steam, can be measured by an indicator. In steam turbines the energy transmitted by the steam to the blades cannot be ascertained in such a manner, so that the shaft horse-power (S.H.P.) ascertained by a brake test or by measuring the torque on the shaft with a torsion meter* is taken as the unit of work for the output.

Up to the present all marine engine data have been referred to in terms of indicated horse-power (I.H.P.), and it is therefore of importance to ascertain the relation between the S.H.P. of a turbine equipment and the I.H.P. of an equivalent reciprocating engine of the usual construction, to enable a comparison of the two installations to be obtained. As the results of careful trials with reciprocating engines at sea there is a large amount of reliable data available for the design of marine turbines, and to allow of these data to be utilised it is essential that the ratio of the S.H.P. of the turbine to the I.H.P. of the equivalent reciprocating engine should be accurately known.

If two exactly similar vessels with propellers of the same efficiency are equipped, the one with reciprocating engines and the other with turbines, the S.H.P. of the turbine will be slightly less than the indicated output of the sister ship.†

It is, however, seldom possible to design a turbine for a speed at which the propeller efficiency will be the same for both types of vessel.

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* See Bauer and Robinson, "Marine Engine and Boilers," p. 599.

† See Bauer and Robinson, p. 3.

In high powered torpedo boats designed to steam at high speeds this is possible. For reasons of economy the speed of the reciprocating engine must be high, whereas the speed of the turbine can be kept comparatively low without exceeding the weight of the reciprocating engine, so that the difference in speed of the propellers will not be so marked as with vessels of other types. The result is that the efficiency of the propellers will be about the same in both vessels, and the S.H.P. of the turbine at a certain speed will be less, or at least no greater, than the I.H.P. of the equivalent reciprocating engine. With small fast cruisers with comparatively large engines designed for high speeds equally favourable results can be obtained for steam turbines.

With slow torpedo boats and ordinary cruisers the effective output of the steam turbine will have to be greater than the I.H.P. to obtain the same speed, and this increase becomes considerable with slower and heavier vessels such as battleships, fast passenger steamers, and the slower speed vessels of the mercantile marine. In all these cases it is assumed that the weight of the turbines is no greater than that of the equivalent engine. If unlimited weight were permissible it would in all cases be possible to run the turbine at a speed enabling the same propeller efficiency as with a reciprocating engine to be obtained, and at the same time to ensure a low steam consumption.

The problem of the future is to devise means for decreasing the weight of the turbine and maintaining its economy. If this can be done it would in most cases be possible to obtain the same propeller efficiency and the same engine efficiency as with a reciprocating engine.

§ 5. Comparative Tests of Effective and Indicated Output.

1. The first test of this nature was made in a series of valuable trials in 1903-04 with two small cruisers of the Imperial German Navy, one the "Hamburg" fitted with reciprocating engines, and the "Lübeck" fitted with Parsons turbines. The results are given in Table 1.

TABLE 1.

	Speed of Vessel.	I.H.P. N_i.	S.H.P. N_s.	Increase of S.H.P. above I.H.P.	Revs. per min.	Ratio of R.P.M.	No. of Shafts.	No. of Propellers.	Displacement. Tons, M.	Weights of Turbines and Boilers.
"Hamburg"	23	11,250	...	22·6%	146	4·3%	2	2	3,220	7 per cent. less in "Lübeck" than in "Hamburg."
"Lübeck"	23	...	13,800		630		4	4	3,210	

2. In the case of a trial with two other cruisers the results were somewhat more favourable to the turbine-driven vessel as the speed of the turbines had been reduced, see Table 2.

TABLE 2.

	Speed of Vessel.	I.H.P. N_i.	S.H.P. N_s.	Increase of S.H.P. above I.H.P.	Revs. per min.	Ratio of R.P.M.	No. of Shafts.	No. of Propellers.	Displacement. Tons, M.	Weight of Turbines and Boilers.
1. Small cruiser with reciprocating engines	24	13,600	...	12·1 %	143	3·69 %	2	2	3,340	The same in both cases.
2. Small cruiser with Parsons' turbine	24	...	15,250		528		4	4	3,410	

The results given in this table are more favourable than in Table 1, but in both cases the number of revolutions of the turbines were chosen too high, with the object, no doubt, of obtaining a low steam consumption at cruising speeds and a saving in weight. Further, the complicated shape of the stern of the boat due to the four stern tubes and brackets required by the Parsons equipment is no doubt responsible for considerable losses.

3. The United States Navy has made comparative trials with the two cruisers "Salem" (Curtis turbine with two shafts each with one propeller) and "Birmingham" (twin screw reciprocating engines), each cruiser having a displacement of about 3,800 tons. The resulting outputs and number of revolutions at various speeds are given in Table 3.

TABLE 3.

	Speed of Vessel.	I.H.P. N_i.	S.H.P. N_s.	Increase of S.H.P. above I.H.P. per cent.	Revs. per min.	Ratio of R.P.M. per cent.	No. of Shafts.	No. of Propellers.	Displacement, Tons, M.
"Birmingham"	24	14,450	...	11	187	1·79	2	2	3,800
"Salem"	24	...	12,850		335		2	2	3,800
"Birmingham"	20	7,250	...	13·8	150	1·86	2	2	3,800
"Salem"	20	...	6,250		249		2	2	3,800
"Birmingham"	16	3,700	...	12·2	120	1·86	2	2	3,800
"Salem"	16	...	3,250		222		2	2	3,800

In this case the indicated output of the reciprocating engine was greater than the effective output of the turbine by about the same amount as the usual difference in I.H.P. and S.H.P. of a reciprocating engine. The propeller efficiency of both engines must have been approximately the same. The reason for this is the comparatively low speed and the simplicity of the stern of the turbine vessel and the comparatively high and therefore unfavourable speed of the reciprocating engine.

4. Similar results have been obtained with torpedo boats driven by A.E.G. turbines, the tests proving that under similar conditions as to speed, shape of the hull, and weight, the S.H.P. output of the turbine vessel was 5 per cent. below the I.H.P. of the reciprocating engine. The proportion of the number of revolutions of the turbine to reciprocating engines was 1·82 or about the same as for the two cruisers in Table 3.

§ 6. Although a knowledge of propeller calculation is necessary to obtain a correct view of the relation of the indicated to effective horsepower of reciprocating engine and turbine-driven vessels (see Part V.), the following can be stated :—

To make the difference between the effective turbine output and the indicated output of an equivalent reciprocating engine small it is necessary that the dimensions of the propeller are as much as possible below the limiting values given in Table 15, *Part V.*

In most cases these proportions of the propellers can only be realised for turbines of high speed vessels of large power where the number of revolutions of the turbine is low and the weight of the turbine the same as the weight of the reciprocating engine.

§ 7. **Example.**—Torpedo boats of 360 tons displacement and varying speed, twin screws assumed. Turbine installation to consist of two A.E.G. turbines.

TABLE 4.

1	2	3	4	5	6
Maximum Speed for which the Vessel is to be built.	Output of Equivalent Reciprocating Engines, I.H.P.	Number of Revolutions per minute for Lightest Design.	Weight of Reciprocating Engine exclusive of Condensing Plant and Boilers in Tons M.	Number of Revolutions per minute of A.E.G. Turbine of Output 2 and Weight 4.	Maximum Revolutions of Propeller to obtain Reasonable Propeller Efficiency.
33	2 × 4,100	310	49	680	860
30	2 × 3,225	325	38	740	800
27	2 × 2,400	340	27·6	825	745
24	2 × 1,650	360	18·8	955	710
21	2 × 1,035	380	11·2	1,150	700
18	2 × 600	400	6·8	1,460	690

This table shows that at 33 knots the permissible number of revolutions is higher than those of the turbine to obtain the same dead weight as the reciprocating engine. It would therefore be possible in this case to save weight by installing turbines. Between 27 and 30 knots the number of revolutions of the turbine is within the number of revolutions required to maintain the given propeller efficiency. Below a certain output and speed (here 28 knots) it is not possible to maintain an economical number of revolutions with a given weight of the turbine, and for a vessel designed to steam at a speed below 28 knots, the turbine will be considerably heavier or will have to develop a larger output than the equivalent reciprocating engine. Under certain conditions a larger boiler plant will be required for the turbine installation, so that in this case, at speeds below 28 knots, the use of turbines as far as weight is concerned will be less favourable. In vessels which are equipped with comparatively heavy reciprocating engines the speed limit at which the turbine becomes heavier than the reciprocating engine will naturally be lower.

SECTION II.

Calculation of the Steam Consumption of the Auxiliaries and the Total Consumption.

§ 8. **General Remarks.**—In designing installations fitted with the reciprocating engine experience alone is the basis on which the main dimensions of the boilers are fixed: experience having indicated how many I.H.P. can be developed with a given type of boiler and engine plant for each unit of heating surface and grate area under forced conditions of steaming.*

This simple method of determining the size of the boiler plant is not advisable in the case of steam turbines. One reason is that, at the time when turbines were first used, they were not built by marine engineers, but by firms who had made a speciality of steam turbine construction for land work and who only supplied the turbines guaranteeing for them a certain steam consumption as is usual in land installations. The main reason, however, is that the calculation of the turbine differs from that of reciprocating engines in that it is required to obtain a definitely determined output of the turbine with a given quantity of steam, so that on estimating a turbine installation the first question will be: "What quantity of steam is available for the turbine?"

§ 9. **Steam Consumption of the Main Turbine and Auxiliaries.**—As the boilers have to supply the steam for the auxiliaries as well as for the main turbine it is important to determine the percentage of the total steam taken by the former, and without a knowledge of this it is impossible to fix in advance the size of the boiler plant for a given installation.

If Q is the total steam consumption of the main turbines, q the steam consumption of the auxiliaries, then the percentage $\frac{q}{Q} \cdot 100$ is approximately that given in Table 5.

* Bauer and Robinson, § 292, p. 474 & ff.

TABLE 5.

Steam Consumption of Auxiliaries in percentage of the Steam Consumption of the Main Turbine.

$$\frac{q}{Q} \cdot 100.$$

Type of Vessel.	Full Speed.	Half Speed.	Slow Speed.
Mercantile - - -	8-10	9-11	10-12
Battleships - - -	10-14	11-14	12-15
Cruisers - - - -	12-15	14-18	15-20
Torpedo boats - - -	16-18	18-20	18-25

The total consumption $Q+q$ for which the boiler plant has to be designed can easily be determined from the above table as soon as the consumption of the main turbines is known. On the other hand, if the available quantity of steam for a given boiler plant is known, the steam available for the main turbines can be found by deducting the steam required by the auxiliaries. The usual ship auxiliaries, such as the electric light machines, winches, hoists, the heating, sanitary and water service pumps, and all the auxiliaries required for the ship's duties, are not included in this table, but only those auxiliaries which are required in connection with the propelling machinery, such as circulating water pumps, air pumps, oil pumps, forced draught and ventilating fans for the stokehold and engine-room, feed pumps, evaporators, in so far as these are required for the boiler make-up feed, and steering engine.

§ 10. As soon as the consumption Q of the main turbines and the consumption q of the auxiliary machinery have been fixed, the boiler plant can be dimensioned from the data given in Bauer and Robinson, "The Marine Steam Engine," Tables 57 and 61.

Table 6 below gives some data of the fuel consumption and evaporation of various types of boilers under forced conditions, and may be of use in determining the size of the boilers.

TABLE 6.

Coal Consumption per sq. m. (sq. ft.) Grate Area and Evaporation per unit weight of Fuel of various Type Boilers.
Feed Water to Boiler at a Temperature of 70° *Cent.* (158° *Fahr.*).

Type of Boiler.	Coal Consumption per Unit Grate Area.		Evaporation. Water at 0° Cent. (32° Fahr.), to Steam at p, Pressure.	Steam Pressure Gauge, p.	
	$\frac{\text{Kg.}}{\text{Sq. M.}}$	$\frac{\text{Lbs.}}{\text{Sq. Ft.}}$		Atm.	Lbs. per Sq. In.
Cylindrical boiler for merchant steamers with natural draught	70-80	2·95-3·38	8-9	14	206
Cylindrical boiler for high-speed merchant steamers, natural draught	90-100	3·8-4·22	8-9	15	220
Cylindrical boiler for high-speed merchant steamers, Howden's forced draught	110-130	4·65-5·5	8·5-9·5	15	220
Belleville boiler for battleships and first-class cruisers; air pressure in stokehold up to 30 mm. ($1\frac{3}{16}$ in.)	120-140	5·07-5·9	8-9	18	265
Schulz-Thornycroft water-tube boiler for battleships; air pressure in stokehold up to 50 mm. (2 in.)	180-200	7·6-8·45	9-10	16-17	235-250
Schulz-Thornycroft water-tube boiler for high-speed cruisers; air pressure in stokehold up to 75 mm. (3 in.)	200-250	8·45-10·55	8·5-9·5	17-18	250-265
Schulz-Thornycroft water-tube boiler for torpedo boats; air pressure in stokehold up to 150 mm. (6 in.)	320-360	13·5-15·2	7·5-8	18-19	265-280
Normand boiler for battleships; air pressure in stokehold up to 50 mm. (2 in.)	180-200	7·6-8·45	7·5-8·5	16-20	235-294
Normand boiler for high-speed cruisers; air pressure in stokehold up to 75 mm. (3 in.)	200-250	8·45-10·55	7·5-8	17-18	250-265
Normand boiler for torpedo boats; air pressure in stokehold up to 150 mm. (6 in.)	350-400	14·7-16·9	7-7·5	18-19	265-280
Yarrow boiler for battleships; air pressure in stokehold up to 40 mm. ($1\frac{5}{8}$ in.)	180-220	7·6-9·3	7·5-8·5	16-17	235-265
Yarrow boiler for high-speed cruisers; air pressure in stokehold up to 60 mm. ($2\frac{3}{8}$ in.)	220-280	9·3-11·8	6·5-7·5	17-18	250-265
Yarrow boiler for torpedo boats; air pressure in stokehold up to 90 mm. ($3\frac{1}{2}$ in.)	400-470	16·9-19·8	6-6·5	18-19	265-280

SECTION III.

REVERSING TURBINES.

§ 11. **General Remarks.**—Up to the present no turbine has been designed allowing of economical reversing, so that special turbines are required for steaming astern. In Part VII. the arrangement of the reversing turbine is described.

In the Curtis and the A.E.G. turbines the reversing turbines are built into the casing of the low-pressure ahead turbines, so that a common exhaust connection is formed for both turbines, and only a single exhaust pipe to the condenser is necessary.

In vessels fitted with Parsons turbines, especially in the quadruple shaft arrangement, high-pressure reversing turbines coupled to the same shaft as the main high-pressure turbines are fitted, in addition to low-pressure reversing turbines, which are built into the same casing as the main low-pressure turbine.

§ 12. **Size of the Reversing Turbine.**—It is naturally advisable that these turbines should occupy as little space as possible, and although in the early days these turbines were usually under-powered, greater care is now given to their design. In spite of this the output developed by the reversing turbines compared to that of the main turbines is considerably less than that of a reciprocating engine when reversed. The latter are capable when reversed of developing as much as 70 to 80 per cent of their normal output, although they are generally not run at more than 50 to 60 per cent.

The following table gives the present usual proportions :—

TABLE 7.

Total Output of the Reversing Turbines as a Percentage of the Total Output of the Main Turbines, and on the Assumption of the same Total Steam Consumption.

Battleships - - - -	40 to 45 per cent.
Small cruisers - - - -	35 „ 40 „
Torpedo boats - - - -	25 „ 30 „

In the case of the Parsons turbines it is difficult to obtain these proportions, as the necessary length to ensure economical expansion of the steam is rarely available. The above table is based on the assumption that the total steam consumption when steaming ahead or astern is the same, so that the efficiency of the reversing turbines will be lower than the efficiency of the main turbines by as many per cent. as the above figures are below 100 per cent.

If, for example, the efficiency of the turbines of a torpedo boat when steaming ahead is 60 per cent., the efficiency when steaming astern will be 15 to 18 per cent.

From the above table it will be seen that the output of the reversing turbines decreases with the size of the vessel. The reason of this is, that to obtain the same ease in manœuvring for turbine vessels as for vessels fitted with reciprocating engine:—

1. The highest obtainable output of a reciprocating engine in small vessels is rarely developed when manœuvring. In such vessels with high-speed engines it is not usual to run the risk of the great strain on the engines by reversing from full speed ahead to full speed astern, so that a certain time elapses before the engine, on being reversed, develops the full output.

2. In small vessels the time occupied in carrying out the necessary manœuvres when reversing is of greater importance than in large vessels. As this requires less time in turbine-driven than engine-driven vessels, the reversing turbines can be smaller in proportion. Due to the absence of the reversing engine in steam turbines the time occupied for reversing a turbine is small, as for the purpose of reversing it is only necessary to close the ahead and open the astern manœuvring valves. This holds good in all cases where each shaft is driven by a single unit. With the reciprocating engine the driver has to watch his engine and manipulate his reversing gear so that the engine stops in the correct position for restarting, this entailing at least one additional operation.

§ 13. **Stopway and Time.**—Frequently the time is specified in which a vessel must stop after the command is given, or alternatively the length is given in which the vessel should come to a standstill. It is difficult to calculate the output of the reversing turbine to fulfil these conditions, and the only thing is to base the output on the results of experiments, of which, however, there are unfortunately few available.

The following may serve to make an approximate calculation :—

The kinetic energy of the vessel in motion after the turbines have been stopped is equal to the sum of the retarding forces acting on the vessel.

If in the following :—

- W be the weight of the ship,
- g acceleration due to gravity,
- N the output of the reversing turbine,
- R the ship resistance,
- v the initial velocity of the ship, *i.e.*, the velocity just before the turbine is stopped,
- η_P the efficiency of the propeller when running in the reversed direction, and
- s the length of the path through which the ship passes after the order "to stop" is given,
- ds the path passed through in an infinitesimal short period of time.

Then—

$$\frac{1}{2}\frac{W}{g}v^2 = \int_0^s \eta_P \, . \, N \, . \, ds + \int_0^s R \, . \, ds$$

PART III.

THE CALCULATION OF STEAM TURBINES.

SECTION I.

General Thermodynamics of Steam Turbine Calculation.

§ 14. **Comparison with the Calculation for Reciprocating Engines.**—Reciprocating engines are calculated on a static basis, whilst the basis for the calculations of steam turbines could be better expressed as dynamic. Consequently, the calculations for reciprocating engines consists in determining the mean pressure of the steam on the piston, so that by multiplying the piston speed by the mean pressure the output of the engine can be obtained. The calculations of a steam turbine depend upon dimensioning the whole and each part of the machine in such a manner that the energy contained in the heat is transformed at a maximum efficiency into mechanical work, and this is only possible through a proper choice of steam and blade velocities in each section of the turbine.

§ 15. **Pressure Volume Diagram.**—In accordance with the method indicated above, the calculation of a reciprocating engine is based upon the alterations in the steam conditions as indicated by the pressure and volume. If in the diagram (Fig. 1) the horizontal lines represent volumes and the vertical lines pressure, and the pressure of the steam is made to vary, whilst the temperature remains constant, the curve connecting the various points will be an isothermal line and of the nature of a hyperbola AB and CD.

The curve connecting the volume and pressure when working without gain or loss of heat is called an adiabatic, such as the lines BC and DA. The area enclosed by these curves represents mechanical work, ABCD being the energy absorbed or given out during one complete cycle. The usual indicator diagram of a reciprocating engine can be represented by such a curve.*

§ 16. **Entropy Temperature Diagram.**—The pressure volume diagram suffers under a disadvantage in so far as the quantity of heat

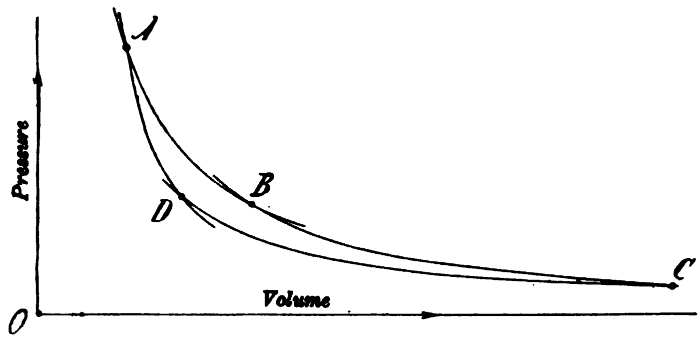

Fig. 1.

involved in the change of the condition of steam as well as the temperature are not directly measurable.

In order to simplify the calculation the well-known entropy diagram (Figs. 2 and 3) is used. In this diagram the quantity of heat is repre-

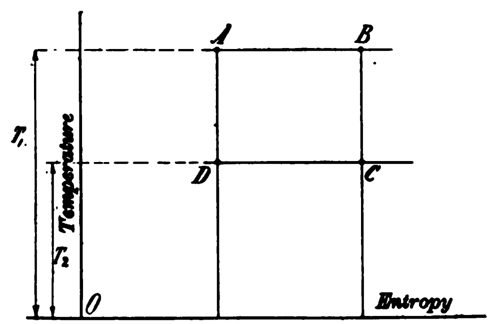

Fig. 2.

sented as an area, the quantity being always that of unit weight of the working fluid (1 kg. or 1 lb.). The ordinates in this diagram represent the absolute temperature of the steam, the abscissæ the integral of the ratio :—

$$\phi = \int \frac{dH}{T},$$

* See Bauer and Robinson, p. 4 & ff.

where $d\text{H}$ is the change in the quantity of heat of the substance for an indefinitely small difference of temperature, and

T the average corresponding absolute temperature.

The integral of this expression is called the Entropy of steam and is usually represented by the letter ϕ.

In the diagram (Fig. 2) the adiabatic expansion of the working substance is shown as a vertical line, as during this period of the cycle work is performed without gain or loss of heat, so that $d\text{H}=0$, in other words the entropy of the working substance remains un-

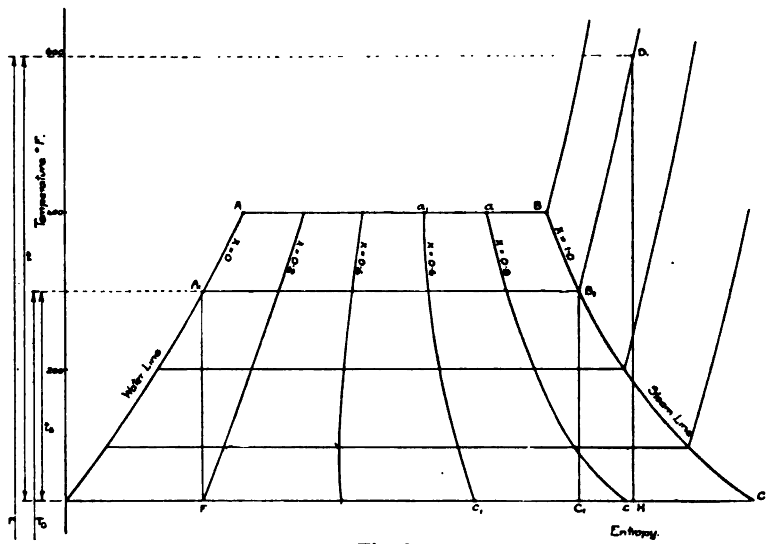

Fig. 3.

altered. The isothermal curve is a horizontal line, as the substance is absorbing heat at a constant temperature in this case, and $\int\frac{d\text{H}}{\text{T}}$ is therefore an increasing quantity. The area ABCD (Fig. 2) represents a cycle between the limits of the temperatures T_1 and T_2 and represents the energy of all the heat which in a perfect heat-engine can be converted into mechanical energy when working between these temperature limits. As the increase in entropy between AB is equal to $\int\frac{d\text{H}}{\text{T}}$ the total quantity of heat absorbed by the substance or the area under AB will be

$$\int\frac{d\text{H}}{\text{T}_1}\text{T}_1=\frac{\text{H}_1\text{T}_1}{\text{T}_1}=\text{H}_1,$$

and the quantity of heat withdrawn from the substance is equal to the area below CD, or

$$\int \frac{dH}{T_2} T_2 = \frac{H_2 T_2}{T_2} = H_2,$$

and as $\frac{H_1}{T_1} = \frac{H_2}{T_2}$, the area ABCD expressed in heat units is equal to

$$\frac{H_1}{T_1}(T_1 - T_2).$$

The heat equivalent of this quantity of heat is the theoretical work which can be obtained by the process.

If the mechanical equivalent of heat is

$$A = \frac{1}{427} \text{ metric units, or}$$

$$A = \frac{1}{778} \text{ English units,}$$

and the mechanical work W in meter kilogram or foot-pounds; then

$$AW = \frac{H_1}{T_1}(T_1 - T_1).$$

In the entropy diagram of steam (Fig. 3), two curves are of special interest: the curve OA representing what is called the water line, and BC the steam line respectively. The curve OA connects the points at which (under the corresponding pressure) steam begins to be formed, and the curve BC the points at which all the water has been evaporated, and when any further addition of heat will result in superheating.

The curve OA_1 B_1 D_1 shows the change in entropy and temperature accompanying the formation of superheated steam at constant pressure from feed water at a temperature 32° Fahr. Up to point A_1 the application of heat results in an increase in both entropy and temperature, but in no change of condition. As soon as the point A_1 is reached, steam begins to be formed and no further change of temperature takes place until at point B_1 where all the water will be in form of vapour or dry saturated steam. Any further application of heat will superheat the steam and increase both the temperature and the entropy of the steam.

Inside the limits of saturated steam the water contained in one unit of weight of the substance is gradually reduced from 100 to 0 per cent. on application of heat, and consequently the quantity of steam increased from 0 to 100 per cent. If x denotes the dryness factor of the steam then at point A_1 $x=0$, and at point B_1 $x=1$. In the diagram (Fig. 3) the curves connecting points of equal dryness factors are shown, so ac for $x = \cdot 8$ and $a_1\ c_1$ for $x = \cdot 6$, &c.

During the period of change in condition of steam * the amount of heat put into the steam corresponds to an increase in the heat of evaporation, *i.e.*, the heat required to raise water from a temperature of 0° Cent. (32° Fahr.) to the required steam condition.

In the following, let:

- H be the total heat of saturated steam per unit weight (1 kg. or 1 lb.),
- h the heat of liquid, *i.e.*, the heat required to raise the temperature of unit weight of water from the temperature of 0° Cent. (32° Fahr.) to saturated steam at a temperature of t_s,
- L the heat of evaporation required to convert one unit weight of water at the temperature t_s to steam of the temperature t_s (latent heat),
- A the mechanical equivalent of heat,
- P pressure in kg. per sq. m.,
- p pressure in kg. per sq. cm. or lbs. per sq. in.,
- v'' the volume of 1 kg. or 1 lb. of saturated steam in cub. m. or cub. ft.,
- v' the volume of 1 kg. of water equal to ·001 cub. m., or the volume of 1 lb. of water equal to ·0167 cub. ft.
- v the volume of 1 kg. or 1 lb. of the mixture of saturated steam and water in cub. m. or cub. ft.,
- x the quantity of steam contained in unit weight of the mixture,

then:

$$H = h + L \text{ heat units,}$$

and as $1 - x$ represents the quantity of water contained in unit weight of the mixture,

$$v = \cdot 001 + x(v'' - \cdot 001),$$

and the total heat of unit weight of the mixture of water and steam

$$H = h + xL.$$

Further, if for unit weight of 1 kg. or 1 lb.,

- H_1 be the total heat of superheated steam,
- h_1 the heat which has to be added to the steam to raise the temperature from t_s to a temperature of superheat of t,
- c_p the specific heat of superheated steam at constant pressure,
- t the final temperature of the superheated steam,
- t_s the temperature of saturated steam at the same pressure,

then the total heat of the superheated steam will be

$$H_1 = h_1 + c_p(t - t_s).$$

* In the following the references are to water and steam only, and not to any other working substance.

As the specific heat of superheated steam is not const creases with the pressure and decreases with the tempera advisable to calculate the total heat from the formulæ (1) belc

To calculate the quantity of heat, the entropy and th volume, the following formulæ in metric measures taken fron "Neue Tabellen und Diagramme für Wasserdampf," can be us

1. The total heat H of dry saturated or H_1 of superheated st

$$H \text{ or } H_1 = 594{\cdot}7 + {\cdot}477t - Xp \text{ cal.}$$

2. The volume v'' of saturated or v_1'' of superheated steam is

$$v'' \text{ or } v_1'' = 47\frac{T}{P} - Y \text{ cub. m.}$$

3. The entropy of saturated or superheated steam is—

$$s = {\cdot}477 \, ln \, T - {\cdot}11 \, ln \, p - Z \, . \, p - 1{\cdot}0544.$$

The value of the constants in the above formulæ are—

$$Y = {\cdot}075\left(\frac{273}{T}\right)^{\frac{10}{3}},$$

$$X = \frac{10{,}000}{427}\left(\frac{13}{3}Y - {\cdot}001\right),$$

$$Z = \frac{10{,}000}{427} \cdot \frac{10}{3} \cdot \frac{Y}{T}.$$

The calculated values of Y, X, and Z, for various temperatures a given in Table 16 at the end of the book.

To arrive at the value necessary for superheating steam the simples method is to deduct from the calculated value of the total heat H the total heat H of saturated steam at the same pressure (taken from Table 16), so that the heat required for superheating is

$$h_1 = H_1 - H.$$

If the steam to be superheated is wet, an additional amount of heat is required for evaporating the water $1 - x$, when raising the steam to the required superheat. The total heat will then be—

$$h_1 = H_1 - H + (1 - x)L.$$

Example 1.—What is the total heat of saturated steam at 20 atm. absolute (294·1 lbs. per sq. in.) and a dryness factor of ·97? (See Steam Table at end of book.)

$h = 215{\cdot}5$	metric units.	$h = 390{\cdot}8$	English units.
$L = 457{\cdot}88$	,,	$L = 813{\cdot}0$	,,
$x = {\cdot}97$	,,	$x = {\cdot}97$	,,

$H = 215{\cdot}5 + {\cdot}97 \, . \, 457{\cdot}88 = 659{\cdot}6$ cal. metric units.

$H = 390{\cdot}8 + {\cdot}97 \, . \, 813 \quad = 1179{\cdot}8$ B.Th.U. English units.

Example 2.—What is the total heat of dry saturated steam of 15 atm. absolute (220 lbs. per sq. in.) pressure?

$h = 200{\cdot}7$	$h = 363{\cdot}4$
$L = 469{\cdot}8$	$L = 836{\cdot}2$
$H = 670{\cdot}5$ cal. metric units.	$H = 1199{\cdot}6$ B.Th.U. English units.

Example 3.—What is the total heat of steam with a dryness factor of ·6 at a pressure of ·1 atm. absolute (1·47 lbs. per sq. in.)?

$h = 45{\cdot}7.$
$x = {\cdot}6.$
$L = 570{\cdot}34.$
$H = h + xL = 45{\cdot}7 + {\cdot}6 \,.\, 570{\cdot}34 = 387{\cdot}9$ cal. metric units.

$h = 82{\cdot}62.$
$x = {\cdot}6.$
$L = 1{,}027{\cdot}2.$
$H = 82{\cdot}62 + {\cdot}6 \,.\, 1027{\cdot}2 = 698{\cdot}94$ B.Th.U. English units.

Example 4.—What is the total heat of superheated steam at a pressure of 15 atm. absolute (220 lbs. per sq. in.) and a temperature of 300° Cent. (572° Fahr.)?

According to formula (1), page 23,

$$h_1'' = 594{\cdot}7 + {\cdot}447t - x \,.\, p.$$

From Table 17 for $t = 300°$, $x = {\cdot}62$.

$$h_1'' = 594{\cdot}7 + {\cdot}477 \,.\, 300 - {\cdot}62 \,.\, 15 = \begin{cases} 728{\cdot}5 \text{ cal. metric units.} \\ 1{,}312 \text{ B.Th.U. English units.} \end{cases}$$

§ 17. **Mollier's Diagram.**—The entropy diagram described in the last paragraph is not directly suitable for the calculation of steam turbines, as it does not allow of simple measurement of the available energy. This could be achieved by drawing curves in the diagram connecting points of constant heat contents, but a more convenient method is that adopted by Dr R. Mollier in his so-called Heat Entropy diagram,* which is now universally used. In this diagram the horizontal distances represent entropy, and the vertical lines the total heat.

When calculating steam turbine only the range immediately above and below the steam line is required, and the diagram is only plotted over this range. During adiabatic expansion entropy is constant. The vertical lines on the diagram are lines of constant entropy, so that a vertical line is the locus of the points representing the condition of steam which is expanding adiabatically. Apart from the

* "Zeitschrift des Vereins Deutscher Inginieure," 1904, p. 271.

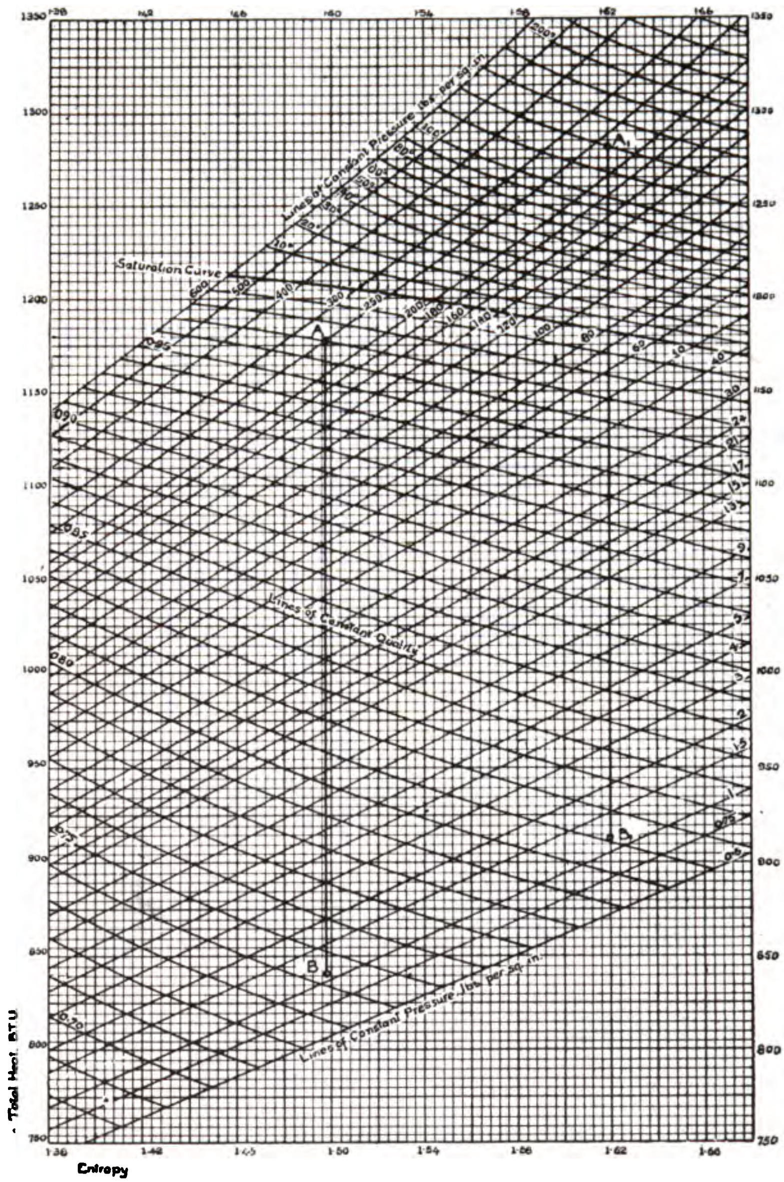

Fig. 4.

losses, steam friction, &c., only adiabatic changes occur in the calculation enabling the heat drop to be measured along a straight line.

The diagram also contains the lines of constant steam quality as well as lines of constant pressure. A copy of this diagram will be found in the pocket at the end of the book.*

A few examples may be useful to show the use of this diagram.

Example 1.—With steam at a pressure of 16 atm. absolute (250 lbs. per sq. in. absolute) and a dryness factor of 97 per cent. expanding adiabatically to a pressure of ·07 atm. absolute (1·1 lbs. per sq. in. absolute), what is the available heat drop for unit weight of steam?

In Fig. 4 a vertical line AB is dropped from the intersection of the curve $p = 250$ lbs. per sq. in., and $x = \cdot 97$ until it meets the line $p = 1 \cdot 1$ lbs. per sq. in.

The total heat at A is $h_a = 1{,}177$ B.Th.U. (657 cal.) and at B $h_b = 840$ B.Th.U. (468·5 cal.). The heat drop, therefore, is $i_a - i_b = 1{,}177 - 840 = 337$ B.Th.U., or in metric measures $657 - 468 \cdot 5 = 188 \cdot 5$ cal. The heat drop could, of course, be measured direct by measuring the distance AB as the scale of the heat in the diagram is constant.

The dryness factor of the steam at the lower pressure is also given in the diagram, and in this case $x = \cdot 75$, the expanded steam containing 25 per cent. water. Generally the following is correct: ***When expanding adiabatically the dryness factor decreases; that is, the moisture in the steam increases.***

Example 2.—With steam at a pressure of 16 atm. absolute (235 lbs. per sq. in.) and a temperature of 280° Cent. (536° Fahr.) expanding to a pressure of ·08 atm. absolute (1·175 lbs. per sq. in.), what is the available quantity of heat contained in 20,000 kg. (44,092 lbs.) of steam?

In Fig. 4 a vertical line is dropped from point A^1, the intersection of the curves $p = 16$ atm. (235 lbs. per sq. in.), and $t = 280°$ Cent. (536° Fahr. or 135° Fahr. superheat), to point B' on curve $p = \cdot 08$ atm. (1·175 lbs. per sq. in.).

The difference of the total heat in point A' and B' is $i_a - i_b = 370$ B.Th.U. (207 cal.). If $h = i_a - i_b$ and Q is the total quantity of steam,

$$Q = 20{,}000 \text{ kg. } (44{,}092 \text{ lbs.});$$

then,

$$H = Q \,.\, h = 20{,}000 \,.\, 207 \text{ cal.} = 4{,}140{,}000 \text{ cal. metric.}$$

$$H = Q \,.\, h = 44{,}092 \,.\, 370 \text{ B.Th.U.} = 16{,}314{,}000 \text{ B.Th.U. English.}$$

* I have to thank Mr Lionel S. Marks and Mr Henry N. Davis for the permission of using the diagram prepared by them for their book, "Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam," published last year by Messrs Longmans, Green & Co. (Transl.).

The dryness factor at B is $x = \cdot 81$, corresponding to a moisture of 19 per cent. in the steam.

Example 3.—What is the final condition of a given quantity of steam at 18 atm. absolute (265 lbs. per sq. in.) admission pressure and a dryness factor of $x = \cdot 98$, when throttled down to a pressure of 5 atm. absolute (73·5 lbs. per sq. in.)?

The process of throttling consists in a reduction of the steam pressure without addition or abstraction of heat. The total heat of the steam will remain unchanged, and the throttling line in the Mollier

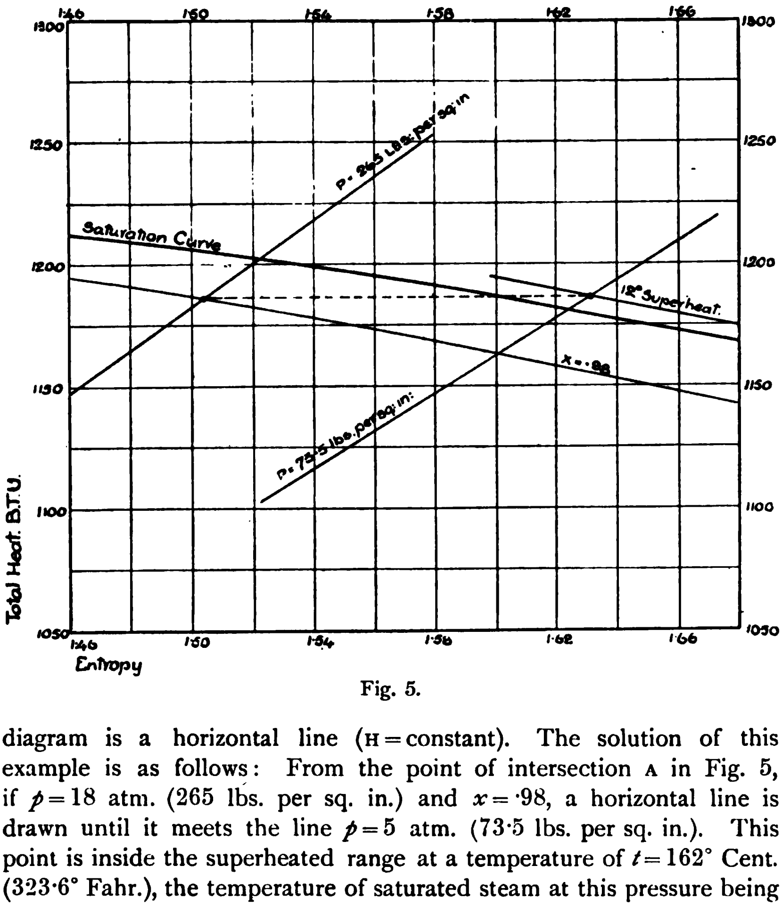

Fig. 5.

diagram is a horizontal line (H = constant). The solution of this example is as follows: From the point of intersection A in Fig. 5, if $p = 18$ atm. (265 lbs. per sq. in.) and $x = \cdot 98$, a horizontal line is drawn until it meets the line $p = 5$ atm. (73·5 lbs. per sq. in.). This point is inside the superheated range at a temperature of $t = 162°$ Cent. (323·6° Fahr.), the temperature of saturated steam at this pressure being $t_s = 151°$ Cent. (304° Fahr.).

From the above it is evident that: Throttling decreases the moisture in the steam, and if the initial condition of the steam is in the neighbourhood of the steam line the effect of throttling can be to superheat the steam.

§ 18. **Calculation of the Efficiency of Steam Turbines.**—The Mollier diagram enables us to calculate the overall efficiency of a known turbine installation.

In the following, let—

p be the admission pressure, *i.e.*, the absolute pressure at which the steam enters the turbine,
x the dryness factor of the steam (see § 23),
t the temperature of the superheated steam,
p_c the exhaust pressure,
N_e the effective output of the turbine,
Q the total steam in kg. or lbs. per hour.

Assuming adiabatic expansion, a knowledge of these data enables the theoretical output of the turbine to be determined under adiabatic expansion.

From the diagram the drop in heat per unit weight of steam is

$$h = h_a - h_e,$$

and the mechanical work in kilogram meter or foot-pounds is

$$W = \frac{Q \cdot h}{A}.$$

The theoretical available output in horse-power will be,

$$N = \frac{Q \cdot h}{A \cdot 3{,}600 \cdot 75} = \cdot 001582\ Q \cdot h = \frac{Qh}{632} \text{ metric units.}$$

$$N = \frac{Q \cdot h}{A \cdot 33{,}000 \cdot 60} = \frac{Qh}{2{,}538} \text{ English units,}$$

and the effective efficiency,

$$\eta_e = \frac{N_e}{N}.$$

See § 45 on the indicated efficiency.

§ 19. **Example.**—For the turbine of a cruiser, let

$Q = 65{,}000$ kg., or 143,300 lbs. per hour,
$p = 17$ atm. absolute, or 250 lbs. per sq. in.,
$p_c = 92$ per cent. vacuum, or 1·179 lbs. per sq. in.
$x = \cdot 97$.

Then the heat drop from the curve will be,

$$h = 186 \text{ cal., or } 335 \text{ B.Th.U.,}$$

and the theoretical output in horse-power,

$$N = \frac{65{,}000 \, . \, 186}{632} = 19{,}100 \text{ H.P. metric units,}$$

or,

$$N = \frac{143{,}300 \, . \, 335}{2{,}538} = 19{,}100 \text{ H.P. English units.}$$

Assuming that the shaft output measured by a torsion meter is 10,800 B.H.P., the effective efficiency will be,

$$\eta_e = \frac{10{,}800}{19{,}100} = \cdot 566.$$

The theoretical output of 19,100 H.P. would correspond to a steam consumption per horse-power hour of

$$\frac{65{,}000}{19{,}100} = 3 \cdot 4 \text{ kg. metric units,}$$

or,

$$\frac{143{,}300}{19{,}100} = 7 \cdot 5 \text{ lbs. English units.}$$

A test on the turbine in question would give a consumption at 10,800 B.H.P. of 6·0 kg. per B.H.P. hour, or 13·23 lbs. per B.H.P. hour.

§ 20. **The Efficiency Ratio.**—Tests have shown that this ratio varies between 65 and 55 per cent., corresponding to a specific steam consumption of about 5·8 to 6·8 kg. (12·8 to 15 lbs.), the lower values being obtained with large turbines and comparatively high speeds, and the higher with small turbines and comparatively low and, therefore, unfavourable speed. The steam consumption of the two systems, A.E.G. and Parsons, is about the same. In both cases a better economy, *i.e.*, a higher efficiency, is obtained with the low-pressure section, and the best means of obtaining a high efficiency of the high-pressure section is in the use of wheels with velocity stages (Curtis), as with these it is possible to obtain a comparatively high efficiency of the high-pressure section inside a minimum of space. There is no doubt that the future development of the steam turbine will be on these lines, and that the marine turbine of the future will be fitted with velocity stages in the high-pressure section, and with a spindle carrying impulse or reaction stages for the low-pressure section.

§ 21. **Influence of Vacuum.**—The influence of vacuum in lowering the steam consumption with an increase and raising the consumption with a decrease of vacuum is considerable. A study of the Entropy diagram will show the magnitude of this influence. If the admission pressure of a turbine is 18 atm. absolute (265 lbs. per sq. in. absolute), the dryness factor ·97, and the vacuum at the exhaust flange 90 per cent. (1·474 lbs. per sq. in.), then the theoretical heat drop will be 182 cal. (327·5 B.Th.U.). If the vacuum is increased to 91 per cent. (1·326 lbs. per sq. in.), the heat drop rises to 184·5 cal. (332 B.Th.U.), an increase of vacuum to 92, 93, or 94 per cent. (1·179, 1·031, and ·8841 lbs. per sq. in.), corresponding to an increase of 338·5, 344·5, 351 B.Th.U.

The steam consumption per B.H.P. drops in inverse proportion to the increase in the available heat drop. If on the above assumption the steam consumption is 6 kg., or 13·23 lbs., per S.H.P. hour at 90 per cent. vacuum, then the steam consumption at the different vacua given above will be:—

For 91	per cent.	vacuum	5·92	kg. per	S.H.P.	hour	(13·04	lbs.	per S.H.P.	hour).	
,, 92	,,	,,	5·81	,,	,,	,,	(12·8	,,	,,	,,	).
,, 93	,,	,,	5·72	,,	,,	,,	(12·6	,,	,,	,,	).
,, 94	,,	,,	5·61	,,	,,	,,	(12·37	,,	,,	,,	).

With the assistance of the Mollier diagram the influence of an increase in vacuum can in each case be ascertained, although it must be remembered that this will only give the theoretical value. The actual influence due to an alteration in the vacuum depends upon the blading arrangement of the low-pressure section, the size and shape of the exhaust passages, and will, therefore, be an individual factor of each machine.

An actual example of the influence of the vacuum on the steam consumption is given in Table 8.

TABLE 8.

Influence of Vacuum on the Steam Consumption of A.E.G. Marine Steam Turbines.

Per Cent. Vacuum	At Approximate 70 per cent. of Normal Output.				At Approximate 31 per cent. of Normal Output.			
	Steam Consumption per S.H.P. Hour.		Variation of Consumption per S.H.P. as Compared with a Vacuum of 90 % per cent.	Variation of Consumption per S.H.P. as Compared with a Vacuum of 90 % in per cent. per per cent. Variation in Vacuum.	Steam Consumption per S.H.P. Hour.		Variation of Consumption per S.H.P. as Compared with a Vacuum of 90 % per cent.	Variation of Consumption per S.H.P. as Compared with a Vacuum of 90 % in per cent. per per cent. Variation in Vacuum.
	Kg.	Lbs.			Kg.	Lbs.		
95	5·60	12·35	− 8·94	−1·79	6·73	14·82	−14·81	−2·96
92	5·94	13·1	− 3·41	−1·71	7·45	16·4	− 5·70	−2·85
90	6·15	13·55	...	±1·68	7·90	17·4	...	±2·76
88	6·35	14·0	+ 3·25	+1·63	8·32	18·31	+ 5·32	+2·66
85	6·64	14·62	+ 7·97	+1·59	8·93	19·65	+13·04	+2·61
80	7·10	15·65	+15·45	+1·55	9·89	21·8	+25·19	+2·52
75	7·55	16·65	+22·76	+1·52	10·74	23·65	+35·95	+2·40
70	7·99	17·6	+29·92	+1·50	11·58	25·5	+46·55	+2·33

From the above figures it is apparent that the proportionate reduction in the steam consumption is in some cases greater than the theoretical figures would indicate, and especially at the lower loads. At these the influence of the vacuum is more marked, as the heat drop due to throttling on admission is considerably smaller than at higher loads, so that an alteration in the vacuum has a much greater effect in proportion to the steam consumption.

As these conditions cannot be ascertained in advance without experimental data, it is necessary that the tests at the works should always include a series of tests at varying vacua, to ascertain what the steam consumption of each turbine will be at different vacua under otherwise stable conditions.

§ 22. **Influence of Superheat.**—The Mollier diagram gives also a means of ascertaining the influence of superheat on the steam consumption. Taking the conditions as above with saturated steam at an admission pressure of 18 atm. (265 lbs. per sq. in.), a dryness factor of ·97, and a vacuum 90 per cent. (1·474 lbs. per sq. in.), the heat available will be 184·5 cal. (332 B.Th.U.). If the steam is superheated to a total temperature of 250° Cent. (482° Fahr.), then under otherwise similar conditions the available heat will be 195 cal. (350

B.Th.U.), and at a total temperature of 300° Cent. (572° Fahr.), 208 cal. (375 B.Th.U.). A proportionate decrease in the steam consumption will follow this increase in the available heat.

With reference to the actual influence of the superheat on the steam consumption of the turbine, similar considerations hold good as in the case of the vacuum, *i.e.*, the magnitude of the influence depends upon the construction of the machine. Table 9 gives some figures from actual test results of an A.E.G. turbine under varying degrees of superheat and load.

TABLE 9.

Influence of Superheat on the Steam Consumption of A.E.G. Steam Turbines.

At Approximate 39 *per cent. of the Normal Output.*

Superheat.		Steam Consumption per S.H.P. hour.		Decrease of Steam Consumption per S.H.P. hour as compared with Dry Saturated Steam.	Decrease of Steam Consumption per S.H.P. hour as compared with Dry Saturated Steam per 10° C. or per 18° Fahr. Superheat.	Superheat required to improve Steam Consumption per S.H.P. hour at 1 per cent.	
° Cent.	° Fahr.	Kg.	Lbs.	Per cent.	Per cent.	° Cent.	° Fahr.
0	0	7·21	15·9	...	...	...	...
20	36	6·83	15·05	5·27	2·61	3·79	6·83
40	72	6·52	14·38	9·57	2·39	4·18	7·52
60	108	6·27	13·82	13·04	2·17	4·61	8·3
80	144	6·06	13·37	15·95	1·99	5·03	9·06
100	180	5·91	13·03	18·03	1·80	5·56	10·0
120	216	5·79	12·73	19·70	1·64	6·10	11·0
At Approximate 21 *per cent. of the Normal Output.*							
0	0	10·01	22·1	...	...	...	...
20	36	9·12	20·1	8·89	4·45	2·25	4·05
40	72	8·58	18·93	14·29	3·57	2·80	5·04
60	108	8·16	18·00	18·48	3·08	3·25	5·85
80	144	7·81	17·21	21·98	2·75	3·64	6·55
100	180	7·53	16·6	24·78	2·48	4·03	7·26
120	216	7·30	16·1	27·07	2·26	4·42	7·95

From these figures it appears that the actual effect of superheating is somewhat different than indicated by theoretical consideration.

In connection with this we would point out that not all types of steam turbines are equally suitable for the use of superheated steam.

No doubt the Parsons turbine with its small clearances in the high-pressure section is not as suitable for highly superheated steam as, for instance, the A.E.G. turbine with the considerably larger clearances in the Curtis wheels of the high-pressure turbine. In the latter type of turbine the temperature of the steam is considerably reduced before entering the first wheel at the outlet of the nozzle, as can be seen from the Mollier diagram. If, for instance, the admission pressure is 18 atm. (265 lbs. per sq. in.), and the temperature of the superheated steam 250° Cent. (482° Fahr.), and the pressure at the outlet of the first nozzle 7 atm. (103 lbs. per sq. in.), then the adiabatic drop in the first stage will be 46 cal. (82·8 B.Th.U.). If the first wheel only absorbs 50 per cent. of this heat, then in accordance with the diagram the steam will be at a temperature of 190° Cent. (374° Fahr.), as compared with the temperature corresponding to the saturated steam of 164° Cent. (327° Fahr.). The superheat of the steam is therefore practically removed.

Although it appears from the above that superheating the steam considerably reduces the steam consumption, it must not be overlooked that fuel is required for superheating the steam. The coal consumption as well as the steam consumption, with and without superheat, must be considered, and as a considerable amount of coal has to be burnt for the purpose, the overall saving in coal is less than the saving in steam. Table No. 10 shows an example. In this case the total saving in coal due to superheating, as compared with saturated steam, is only 9·2 per cent., whereas the saving in steam amounts to 21·1 per cent.

TABLE 10.

Estimated Saving in Fuel due to the use of Superheated Steam.

Dry Saturated Steam.			Superheated Steam.		
1. Admission pressure, p -	16 atm. abs.	228 lbs. per sq. in.	1. Admission pressure, p -	16 atm. abs.	228 lbs. per sq. in.
2. Dryness factor, x - -	·96	·96	2. Total steam temperature, t	300° Cent.	572° Fahr.
			3. Superheat	100° Cent.	180° Fahr.
3. Output of turbine, S.H.P.	7,900 H.P.	7,900	4. Output of turbine, S.H.P.	7,900	7,900
4. Steam consumption per S.H.P. hour	7·10 kg.	15·65 lbs.	5. Steam consumption per S.H.P. hour	5·60 kg.	12·35 lbs.
			6. Reduction of steam consumption due to superheat above dry saturated steam	21·1 per cent.	21·1 per cent.
5. Total heat per unit weight of steam at a feed water temperature of 80° Cent., or 176° Fahr.	571·6 cal.	1,030 B.Th.U.	7. Total heat per unit weight of steam at a feed water temperature of 80° Cent., or 176° Fahr. - - - -	647·6 cal.	1,166 B.Th.U.
6. Heat required per S.H.P. hour	4,058 cal.	7,300 B.Th.U.	8. Heat required per S.H.P. hour	3,627 cal.	6,528 B.Th.U.

TABLE 10—(*continued*).

Dry Saturated Steam.			Superheated Steam.		
7. Total steam consumption per hour, 3×4	56,090 kg.	123,657 lbs.	9. Total steam consumption per hour, 4×5	44,240 kg.	97,532 lbs.
8. Evaporation of boiler, kg. or lbs. of water per kg. or lb. of coal	8·7	8·7	10. Evaporation of boiler, kg. or lbs. of water per kg. or lb. of coal	8·7	8·7
			11. Coal consumption per hour, $\frac{9}{10}$	5,085 kg.	11,214 lbs.
			12. Calorific value of coal	7,000 cal.	12,600 B.Th.U.
			13. Coal consumption for superheating per hour (direct fired superheater at 62 per cent. efficiency)	775 kg.	1,710 lbs.
9. Total coal consumption per hour, $\frac{7}{8}$	6,447 kg.	14,213	14. Total coal consumption per hour, $11+13$	5,860 kg.	12,924 lbs.
10. Coal consumption exclusive of auxiliaries per S.H.P. hour	·8161 kg.	1·8 lbs.	15. Coal consumption exclusive of auxiliaries per S.H.P. hour	·741 kg.	1·635 lbs.
			16. Saving in coal due to superheating	92 per cent.	92 per cent.

§ 23. **The Influence of Wet Steam on the Efficiency of the Steam Turbine.**—The dryness factor of the steam depends largely on the type of boilers, the forcing of the boilers, the lagging of the steam range as well as the skill of the stokers. Small steam storage capacity of the boilers, inefficient lagging of the steam pipes, all tend to increase the percentage of moisture of the steam. It is difficult to ascertain the influence of forcing a boiler on the dryness of the steam. With heavily forced boilers the quantity of water carried over by the steam may be considerable; against this has to be set the greatly increased condensation losses in the pipe line when only a small quantity of steam is passing to the turbines. It is generally assumed that with the type of water-tube boilers commonly used in naval vessels the dryness factor of the steam at full speed is between ·97 and ·95 and about ·96 to ·93 at reduced speed.

In the calculations an allowance is made for the dryness factors by subtracting from the available quantity of steam the amount of water carried over from the boilers. If, for instance, 60,000 kg. (132,000 lbs.) of steam are available from the boilers the turbine calculation is based on approximately 57,000 to 58,200 kg. (125,600 to 128,000 lbs.). (See Bauer and Robinson, "Marine Engines and Boilers," p. 590.)

§ 24. **Theoretical Outlet Velocity.**—If a unit weight of steam flows from a nozzle with a velocity of c_0 the kinetic energy contained in the steam is:

$$\mathrm{W} = \frac{c_0^2}{2g},$$

and the velocity will be

$$c_0 = \sqrt{2g\mathrm{W}}.$$

In calculating steam turbines it is usual to start with the available heat drop. If $\mathrm{A} = \frac{1}{427}$ in metric units, or $\frac{1}{778}$ in English units is the mechanical of equivalent heat, then the heat drop will be:

$$h = \mathrm{AW}, \text{ and}$$

$$\mathrm{W} = \frac{h}{\mathrm{A}}.$$

From this

$$c_0 = \sqrt{\frac{2gh}{\mathrm{A}}} = \sqrt{2 \,.\, 9{\cdot}81 \,.\, 427 \,.\, h} \text{ in metric units}$$

$$= \sqrt{2 \,.\, 32{\cdot}2 \,.\, 778 \,.\, h} \text{ in English units,}$$

or,

$$c_0 = 91{\cdot}5 \sqrt{h} \text{ in metric units,}$$

and

$$= 223{\cdot}5 \sqrt{h} \text{ in English units.}$$

Example.—What is the velocity with which saturated steam at a pressure of 18 atm. absolute (265 lbs. per sq. in.) and a dryness factor of ·97 flows into a space in which there is a pressure of 9 atm. absolute (132 lbs. per sq. in.)?

From the Mollier diagram the heat drop at adiabatic expansion is $h = 31$ cal. (55·8 B.Th.U.), and the steam velocity will therefore be:

$$c_0 = 91{\cdot}5\sqrt{31} = 509 \text{ m. per sec. metric units,}$$

or,

$$c_0 = 223{\cdot}5\sqrt{55{\cdot}8} = 1{,}670 \text{ ft. per sec. English units.}$$

Calculation of the Nozzles.

§ 25. **Type of Nozzle.**—Two different types of steam nozzles are used in turbines: parallel nozzles (Fig. 6) and expanding nozzles (Fig. 8). The former are used where the ratio of the initial pressure to the pressure in the space into which the nozzle discharges is less than 1·73, and the second type where this ratio is higher than the above value. The reason for this is to be found in the so-called *critical pressure* (see § 27).

§ 26. **Calculation of Parallel Nozzles.**—Let

F_a be the outlet area of the nozzle measured at right angles to the axis of the nozzle,
Q the quantity of steam passing through the nozzle,
v the specific volume of the steam at the outlet,
c_1 the steam velocity at the outlet.

Then at any moment the following holds good:

$$Qv = F_a \cdot c_1.$$

The velocity c_1 is the actual outlet velocity and is always lower than the theoretical velocity c_0. If the latter is obtained from the entropy diagram equal to

$$c_0 = 91{\cdot}5\sqrt{h} \text{ metric units,}$$
$$c_0 = 223{\cdot}5\sqrt{h} \text{ English units,}$$

then

$$c_1 = \phi c_0.$$

The value of the constant ϕ varies between ·85 to ·97 depending on the construction and workmanship of the nozzle. The final condition of the steam at the outlet from the nozzle can be determined from the entropy diagram as follows:—

If in Fig. 6 α is the angle of the nozzle axis with the direction of the wheel velocity, then the outlet area of the nozzle at right angles to the nozzle ring will be

$$F_a' = \frac{F_a}{\sin \alpha}.$$

If the quantity of steam is:

$$Q = 36{,}000 \text{ kg. per hour, or say } 80{,}000 \text{ lbs. per hour,}$$

$$Q_s = \frac{36{,}000}{3{,}600} = 10 \text{ kg., or say } \frac{80{,}000}{3{,}600} = 22{\cdot}2 \text{ lbs. per second,}$$

and further

the initial pressure 16 atm. absolute, or say 235 lbs. per sq. in.
the outlet pressure 10 atm. absolute, or say 147 lbs. per sq. in.
the initial dryness factor ·97, ·97,

then the ratio of the initial pressure to the final pressure is 1·6, that is, the pressure at the outlet of the nozzle is higher than the critical pressure corresponding to the initial pressure, and consequently expanding nozzles cannot be used.

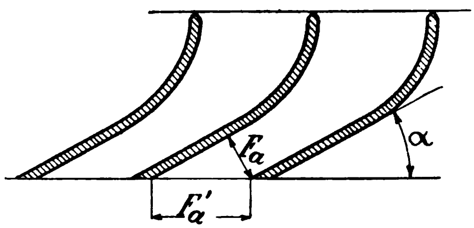

Fig. 6.

If the velocity factor $\phi = {\cdot}9$ then we obtain an available heat drop from the entropy diagram of $h = 21$ cal. or 38 B.Th.U. As the actual velocity is only 90 per cent. of the theoretical velocity, and as the loss in energy is proportional to the differences of the squares of the velocities, the loss will be:

$$1^2 - {\cdot}9^2 = 1 - {\cdot}81 = {\cdot}19, \text{ say 20 per cent.}$$

To obtain the final condition of the steam we mark off on the vertical AB equal to h a point C, so that BC = 20 per cent. of AB (see Fig. 7). AC then represents the actual heat drop equal to 16·8 cal. or 31·7 B.Th.U.

The final condition of the steam is obtained by drawing a horizontal line through C till it meets the pressure curve corresponding to 10 atm. or 147 lbs. per sq. in. at the point D corresponding to a dryness factor of ·947. This indicates that the loss in velocity appears as a gain in heat.

The specific volume can be obtained from the tables of the properties of saturated steam. The specific volume of dry saturated steam at a pressure of 10 atm. is $v_s = \cdot 199$, or at 147 lbs. per sq. in. $v_s = 3\cdot 072$, and for wet steam at $x = \cdot 947$.

$$v = \cdot 199 \,.\, \cdot 947 = \cdot 189 \text{ cub. m. metric units, or}$$
$$v = 3\cdot 072 \,.\, \cdot 947 = 2\cdot 906 \text{ cub. ft. English units.}$$

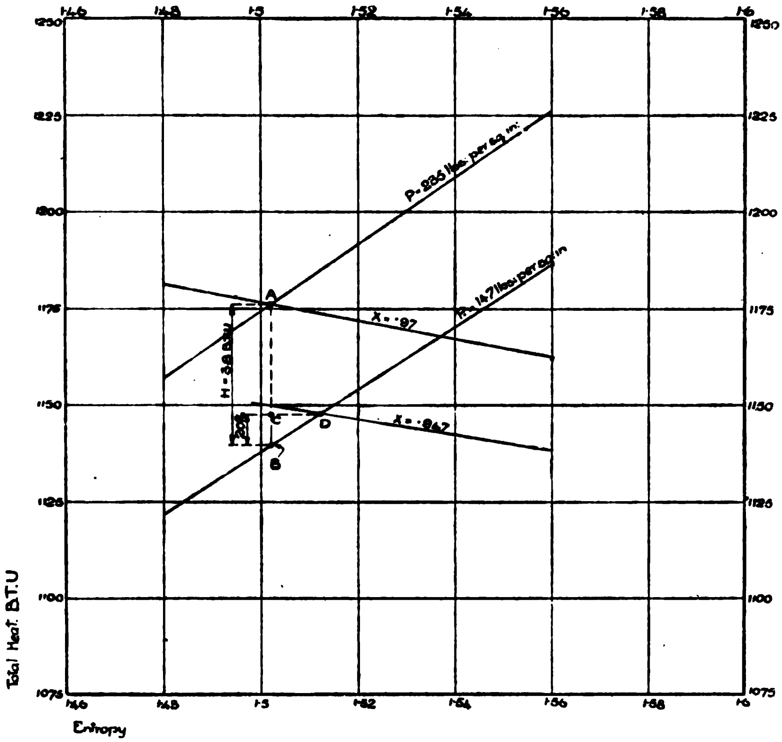

Fig. 7.

The theoretical velocity can be obtained from

$$c_0 = 91\cdot 5 \sqrt{h} = 91\cdot 5 \sqrt{21} = 419 \text{ m. per sec. metric units,}$$
$$c_0 = 223\cdot 5 \sqrt{h} = 223\cdot 5 \sqrt{38} = 1{,}374 \text{ ft. per sec. English units,}$$

and the actual outlet velocity,

$$c_1 = \cdot 9 \,.\, 419 = 377 \text{ m. per sec. metric units,}$$
$$c_1 = \cdot 9 \,.\, 1{,}374 = 1{,}237 \text{ ft. per sec. English units,}$$

and,

$$F_a = \frac{Q \,.\, v}{c_1} = \frac{10 \,.\, 0\cdot 189}{377} = \cdot 005 \text{ sq. m.} = 50 \text{ sq. cm. metric units,}$$

$$F_a = \frac{Q \,.\, v}{c_1} = \frac{22\cdot 2 \,.\, 2\cdot 906}{1{,}237} = \cdot 0522 \text{ sq. ft.} = 7\cdot 52 \text{ sq. in. English units.}$$

Assuming that $\alpha = 30°$, then,

$$F_a' = \frac{F_a}{\sin \alpha} = 2F_a = 100 \text{ sq. cm. metric units,}$$
$$= 15{\cdot}04 \text{ sq. in. English units.}$$

The blade thickness, pitch and width of the nozzle is determined from a knowledge of practical designs so that the length of the arc over which admission takes place can be calculated. If the width of the nozzle is 3·5 cm. (1·377 in.), the pitch 3 cm. (1·18 in.), and the blade thickness ·15 cm. (·059 in.), then the developed length of the arc will be:

$$l = \frac{100}{3{\cdot}5} + \left(\frac{l}{3{\cdot}0} - 1\right) {\cdot}15,$$

or $l = 30$ cm. (11·8 in.) requiring ten nozzles.

Calculation of Expanding Nozzles.

§ 27. **Critical Pressure.**—If steam flows through a cylindrical orifice with sharp or slightly rounded edges it can be shown by calculation, and has been proved experimentally, that the velocity of the steam cannot exceed a certain amount, even if the ratio of the inlet and outlet pressure is made as small as possible.

This occurs when the ratio of the two pressures exceeds 1 : ·577, *i.e.*, if the critical pressure is exceeded.

If the initial pressure is p_1, then the critical pressure will be

$$p_2 = {\cdot}577 \, . \, p_1.$$

If the pressure at the outlet p_2 drops below this value no further increase in the velocity of the steam will result.

On the other hand, if the pressure at the outlet remains p_2, then no increase in the steam velocity takes place if the initial pressure exceeds

$$p_1 = \frac{p_2}{{\cdot}577} = 1{\cdot}73 p_2.$$

§ 28. **Calculation of the Critical Pressure and the corresponding Outlet Velocity.***

Let, in the following:—

F be the area of the nozzle in sq. m. or sq. ft.,

Q_s the quantity of steam in kg. per sec. or lbs. per sec.,

v the specific volume of the steam at the outlet in cub. m. or cub. ft.,

w the outlet velocity in m. per sec. or ft. per sec.

* See Zeuner, "Turbinen," 1899, pp. 268 and ff.

Then the following formula holds good :—

$$Q_s \cdot v = F \cdot w.$$

The equation for saturated steam is :

$$p \cdot v^n = p_1 \cdot v_1^n.$$

From this formula the following formula for the outlet velocity is deducted :

$$w = \sqrt{2g \frac{n}{n-1} p_1 v_1 \left(1 - \frac{p}{p_1}\right)^{\frac{n-1}{n}}},$$

and for the quantity of steam from the nozzle per second,

$$Q_s = F\frac{w}{v} = F\sqrt{2g\frac{n}{n-1} \cdot \frac{p_1}{v_1}\left[\left(\frac{p}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p}{p_1}\right)^{\frac{n+1}{n}}\right]}.$$

Q_s becomes a maximum when :

$$p = p_m = p_1\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}},$$

and for a pressure of $p = p_m$,

$$Q_s = F_m\sqrt{2g\frac{n}{n-1} \cdot \frac{p_1(n-1)}{v_1(n+1)}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}},$$

and

$$w_m = \sqrt{2g\frac{n}{n+1} \cdot p_1 v_1}.$$

For dry saturated steam the index n in the adiabatic equation is

$$n = 1{\cdot}135,$$

so that

$p_m = {\cdot}5744 p_1$ metric units,

$w_m = 323 \sqrt{p_1 v_1}$ „

$Q_s = 200\ F_m\sqrt{\frac{p_1}{v_1}}$ „

or,

$p_m = {\cdot}5744 p_1$ English units.

$w_m = 70{\cdot}2 \sqrt{p_1 v_1}$ „

$Q_s = 43{\cdot}25\ F_m\sqrt{\frac{p_1}{v_1}}$ „

For F_m in sq. mm. and Q_h (per hour),

$Q_h = {\cdot}72\ F_m\sqrt{\frac{p_1}{v_1}}$ metric units, or with F_m in sq. in.

$$Q_h = 14{,}370\ F_m\sqrt{\frac{p_1}{v_1}}\ \text{English units.}$$

The above formulæ hold good for the outlet opening of a parallel nozzle. In an expanding nozzle the velocity of the steam increases above the velocity w_m up to the value corresponding to the heat drop due to the inlet pressure and outlet pressure, so that the theoretical velocity at the outlet of the expanding nozzle is

$$c_o = 91{\cdot}5\ \sqrt{h}\ \text{metric units, or}\ c_o = 223{\cdot}5\ \sqrt{h}\ \text{in English units,}$$

exactly as if the pressure had not exceeded the critical value.

From the above it is evident that if the pressures in the two spaces connected by the nozzle exceed the critical pressure, the nozzle will have to be fitted with diverging or expanding outlet.

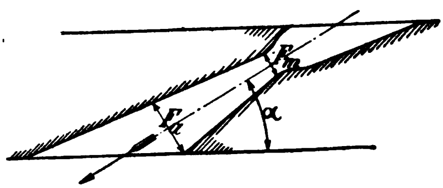

Fig. 8.

Such a nozzle is shown in Fig. 8. From the above formulæ, the opening at the neck of the nozzle will be

$$F_m = \frac{Q_h}{{\cdot}72\sqrt{\frac{p_1}{v_1}}}\ \text{metric units,}$$

$$F_m = \frac{Q_h}{14{,}370\sqrt{\frac{p_1}{v_1}}}\ \text{English units.}$$

The area at the outlet is :—

$$F_a = Q_s \cdot \frac{v_a}{w_1},$$

where $w_1 = \phi w_o$, the actual outlet velocity, and v_a the specific volume of the steam at the outlet.

As a nozzle of this type is capable of reducing the initial pressure to a low pressure they are suitably used in the main turbines, as it is possible to reduce the steam at boiler pressure to a low pressure at the first stage.

The following example will serve to show the advantage gained by the use of this type of nozzle:—

§ 29. **Example of the Calculation of an Expanding Nozzle.**—Let, in the following:—

p_1 be the initial pressure at 15 atm. absolute, or 221 lbs. per sq. in. absolute.

p_2 „ outlet pressure at 2 atm. absolute, or 29·4 lbs. per sq. in. absolute.

Q_h „ quantity of steam flowing through the nozzle in kg. or lbs. per hour at 720 kg. or 1,590 lbs.

Q „ quantity of steam flowing through the nozzle per second at ·2 kg. or ·442 lb.

v_1 „ specific volume of the steam at the inlet at ·136 cub. m. or 2·082 cub. ft.

v_a „ specific volume of the steam at the outlet of the nozzle.

On the assumption of dry saturated steam, the correction A_m at the throat of the nozzle is:

$$F_m = \frac{Q_h}{\cdot 72\sqrt{\frac{p_1}{v_1}}} \text{ metric units,}$$

$$F_m = \frac{720}{\cdot 72\sqrt{\frac{15}{\cdot 136}}} = 68\cdot 4 \text{ sq. mm. metric units,}$$

or,
$$F_m = \frac{1{,}590}{14{,}370\sqrt{\frac{221}{2\cdot 082}}} = \cdot 1072 \text{ sq. in. English units.}$$

The outlet cross-section is:

$$F_a = \frac{Q_s \cdot v_a}{c_1}.$$

If $\phi = \cdot 9$ $\therefore c_1 = \cdot 9 \,.\, 91\cdot 5\sqrt{h} = 760$ m. per sec. metric units,
$= \cdot 9 \,.\, 223\cdot 5\sqrt{h} = 2{,}490$ ft. per sec. English units,
h being obtained from the entropy diagram as 85 cal. or 153 B.Th.U.

The entropy diagram can also be used to determine v_a. On the vertical line corresponding to the heat drop in the diagram, a length equal to 20 per cent. of the total heat drop, corresponding to a loss in velocity of 10 per cent. (see § 26), is cut off and a horizontal line drawn from this point until it cuts the pressure curve $p_2 = 2$ atm., or 29·4 lbs.

per sq. in. This point gives the condition of the steam at the outlet, and based on the above data this becomes $x = \cdot 915$, and the specific volume will be:

$$v_a = \cdot 915 \,.\, \cdot 901 = \cdot 824 \text{ cub. m.,}$$
$$v_a = \cdot 915 \,.\, 13 \cdot 92 = 12 \cdot 75 \text{ cub. ft.,}$$

·901 cub. m. or 13·92 cub. ft. being the specific volume of dry saturated steam at the pressure p_2.

Then the outlet cross-section will be:

$$F_a = \frac{\cdot 2 \,.\, \cdot 824}{760} = 217 \text{ sq. mm.,}$$

or,

$$F_a = \frac{\cdot 442 \,.\, 12 \cdot 75}{2{,}490} = 3 \cdot 26 \text{ sq. in.}$$

Assuming constant width of the nozzle the shape will be as shown in Fig. 8.

If an expanding nozzle had not been used the outlet velocity would have been

$$w_m = 323 \sqrt{p_1 v_1} = 323 \sqrt{15 \,.\, \cdot 136} = 462 \text{ m. per sec.,}$$

or,

$$w_m = 70 \cdot 2 \sqrt{221 \,.\, 2 \cdot 082} = 1{,}506 \text{ ft. per sec.,}$$

in place of 844 m. per sec. or 2,760 ft. per sec.

This value could also be obtained in a different manner, as in a parallel nozzle the steam velocity cannot be higher than that corresponding to the critical pressure. The latter in the above case is:

$$p_m = \frac{15}{1 \cdot 73} = 8 \cdot 7 \text{ atm. absolute,}$$

or,

$$p_m = \frac{221}{1 \cdot 73} = 128 \text{ lbs. per sq. in.}$$

The heat drop between 15 and 8·7 atm., or 221 and 128 lbs. per sq. in., is 25·5 cal. or 45·4 B.Th.U., and the corresponding velocity,

$$w_m = 91 \cdot 5 \sqrt{25 \cdot 5} = 462 \text{ m. per sec.,}$$

or,

$$w_m = 223 \cdot 5 \sqrt{45 \cdot 4} = 1{,}506 \text{ ft. per sec.}$$

This would indicate that the difference between the heat drop of 85 cal. and 25·5 cal., that is, 59·5 cal. or 153 B.Th.U. and 46 B.Th.U., is lost. Experiments show, however, that this is not the case in a turbine, as the steam continues to expand in the clearance and rotating blades after issuing from the nozzle at the critical pressure. This results in an increase of velocity above that due to the critical pressure.

SECTION II.

GENERAL CONDITIONS FOR THE CALCULATION AND DESIGN OF STEAM TURBINES.

§ 30. **Definition of a Steam Turbine.**—The steam turbine is a prime mover in which the energy contained in the steam is transformed into mechanical work by withdrawing from the steam the kinetic energy in allowing it to flow along moving curved blades.

§ 31. **Comparison with Water Turbines.**—The main difference between steam and water turbines is the difference in the working fluid and the consequent difference in the construction. The volume of the water passing through a turbine remains unaltered, whereas steam is a fluid which tends to increase in volume immediately the pressure acting on it is decreased.

§ 32. **Reaction and Impulse Turbines.**—In the same manner as with water turbines there are two main types of steam turbines:

1. Reaction turbines, and
2. Impulse turbines.

Definition of a Reaction Turbine.—The simplest type of reaction turbine would be a chamber which could rotate round an axis and to the interior of which steam were admitted. If nozzles were fitted to this chamber so that the steam could issue from the chamber in a tangential direction the impulse given by the steam would cause the chamber to rotate. In this case the whole expansion of the steam takes place in the nozzles of the rotating chamber. Based upon this primitive type of turbine, reaction blading is such in which the area between two adjacent blades decreases towards the outlet, so that the velocity of the steam is greater when leaving than when entering the blades.

Pure reaction type turbines in which the whole expansion occurs in the moving blades have never been practically used. The known types have all a degree of reaction of about 50 per cent., *i.e.*, about half the heat drop is converted into kinetic energy in the fixed and the other half in the rotating blades, *i.e.*, the steam expands in the fixed as well as in the rotating blades.

Definition of the Impulse Turbine.—In an impulse turbine the steam after passing through a fixed nozzle flows through blades fixed on the circumference of a rotating wheel, in such a manner that the jet of steam enters the moving blades with as small a shock as possible, so as to ensure a gradual change in the direction of flow and to maintain a uniform velocity of the jet relatively to the blades. For this to be the case it is necessary that the width between adjacent blades is constant.

Unlike the reaction turbine the whole expansion of the steam takes place in the fixed blades, no change of volume or pressure occurring within the moving blades (exclusive, of course, of losses due to friction, &c.). The impulse blades are of the type shown in Fig. 9.

§ 33. As the conversion of the heat energy in the steam into kinetic energy depends upon changes in the direction of flow of the steam jet moving at a given velocity, the calculation of steam turbines is based upon the combination of the steam velocities with the blade velocity by means of vector diagrams.

For combining velocities the same theorem holds good as for the combining of forces, *i.e.*, the resultant of two velocities is the diagonal of the parallelogram formed by the two velocities.

Example.—Assume a turbine blade to be fixed at the circumference of a wheel of the diameter D (D being the mean blade ring diameter from the centre to the centre of the blades) rotating at n revolutions per minute. The blade velocity per second will be:

$$u = \frac{D\pi n}{60}.$$

If a jet of steam at the velocity c_1 impinges on the blades at an angle equal α, the velocity with which the steam moves along the blades and relatively to the blades can be found from the construction of the velocity diagram given in Fig. 9.

Starting from a point A we make AB in direction and magnitude equal to c_1. In point B the velocity u is added in an opposite direction to that of rotation, so that $BC = u$. Then the required direction and magnitude of the relative velocity is AC. With this velocity $AC = w_1$, the steam flows relatively along the blades on the assumption that there is no retarding action due to friction.

The absolute velocity with which the steam leaves the blades can be ascertained in the same manner. If α_1 is the angle at which the steam leaves the blade, then in the case of impulse blading (neglecting friction) the relative outlet velocity w_2 at the outlet is equal to the inlet velocity

w_1 to the blades, so that DE $= w_2 = w_1$. By combining this velocity w_2 with the blade velocity u so that EF $= u$, the resultant DF $= c_2$ gives the required absolute outlet velocity. The reason for taking the velocity u in the first case in opposite direction to the blade velocity, and in the second case in the same direction, is, that in the first case the relative velocity must have a tangential component smaller by the amount u than the absolute outlet velocity c_1, whereas in the second case the absolute outlet velocity c_2 must have a tangential component smaller by the amount u than the relative outlet velocity w_2.

§ 34. **The Fundamental Formulæ for the Work in a Turbine.**—In accordance with the definition given in § 30 the steam in a turbine does work by flowing along the curved blades, maintaining them

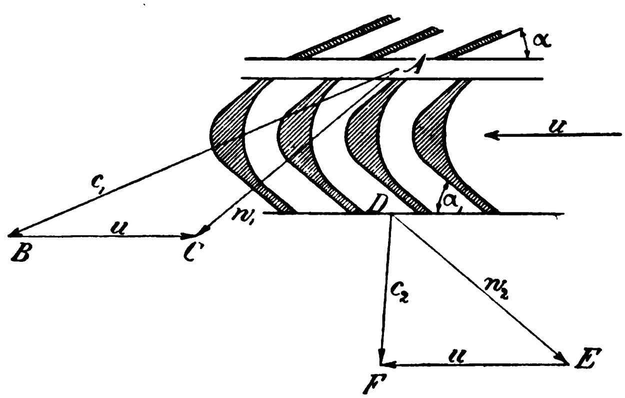

Fig. 9.

in motion against a resistance. Accordingly the work done by the steam on each blade depends upon the change in the absolute velocity.

According to the fundamental formula of mechanics—

$$\text{Force} = \text{Mass} \times \text{Acceleration},$$

or,

$$P = m \cdot \frac{dv}{dt},$$

and

$$P \cdot dt = m \cdot dv;$$

further,

$$\int_{t_1}^{t_2} P \cdot dt = \int_{v_1}^{v_2} m \cdot dv,$$

from which formulæ we arrive at the following:

$$P(t_1 - t_2) = m \cdot (v_1 - v_2),$$

or for unit of time,

$$P = m(v_1 - v_2) = \frac{Q}{g} \cdot (v_1 - v_2).$$

Where

$v_1 - v_2$ is the change in the absolute velocity of the steam per second,

Q the quantity of steam passed per second,

g acceleration due to gravity.

The work done per unit of time is the product of the force P acting in a tangential direction into the path per unit of time (u).

We have, therefore,

$$W = u \cdot P = u\frac{Q}{g}(v_1 - v_2).$$

The velocities v_1 and v_2 represent the tangential components of the steam velocities as these alone are responsible for the work done in the wheel, which can only move in the direction of the tangent.

From this fundamental formula most of the data of the blading can be deducted.

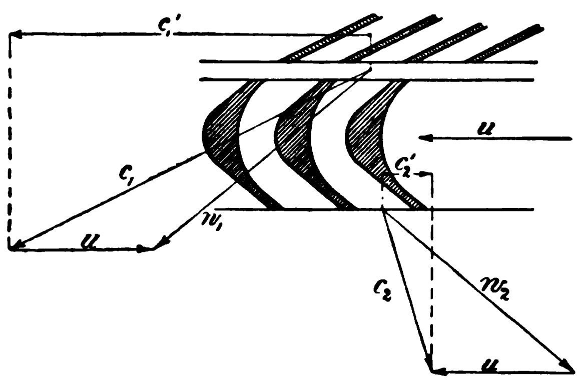

Fig. 10.

§ 35. **Determination of the Blade Efficiency from the Blade Angles.**—The energy imparted to the blades by the steam is equal to the tangential force multiplied by the path covered in unit of time, *i.e.*, the blade velocity. If the absolute steam velocity at the outlet of a nozzle or a fixed blade is c_1 (Fig. 10), the absolute velocity at the outlet of the rotating blades c_2, then the tangential components of these velocities are c_1' and c_2'. The retardation of the steam in a tangential direction will be $c_1' - c_2'$, and if Q is the quantity of steam flowing through the blades per second, g the acceleration due to gravity, then the tangential force will be—

$$\frac{Q}{g}(c_1' - c_2').$$

If u is the tangential velocity of the blades the energy transmitted to the wheel will be—

$$W = u\frac{Q}{g}(c_1' - c_2'),$$

or the mechanical equivalent of heat per unit of weight,

$$h_i = \frac{Au}{g}(c_1' - c_2').$$

If h denotes the available heat drop of the steam between the inlet to the nozzle and the outlet from the moving blades, the efficiency will be:

$$\eta = \frac{h_i}{h} = \frac{Au}{gh}(c_1' - c_2').$$

If the velocities c_1' and c_2' oppose one another as in Fig. 10, *i.e.*, if one has a component in the direction of rotation and the other one in an opposite direction, then the tangential components have to be added and not subtracted, so that in each case both magnitude and direction must be taken into account.

§ 36. **Ideal Impulse Turbine.**—As stated in § 32 the fundamental type of an impulse turbine consists of rotating blades on which impinges the steam issuing from a nozzle. The whole of the energy contained in the steam on leaving the nozzle is converted into velocity, this velocity being transformed into mechanical work in the wheel. Fig. 11 represents a nozzle and one blade of such a turbine.

We will first of all assume the simplest case, that is, angles of $\alpha_1 = 0$ and $\alpha_2 = 180°$.

The energy imparted to the wheel by the steam jet per second is:

$$W = \frac{Q}{g} . u . (v_1 - v_2).$$

In this case the velocity v_1 is equal to the absolute velocity of the steam jet which we denote with c_o and the absolute final velocity v_2 with c_2.

A simple reflection will show that this final velocity c_2 is equal to the absolute inlet velocity c_o less twice the blade velocity.

Or, $$c_2 = c_o - 2u.$$

The relative velocity of the steam in the blade is:

$$w_1 = c_o - u.$$

The absolute outlet velocity must be less than the relative velocity in the blades by the amount u as the blade is moving through space with this velocity in an opposite direction. The total change in velocity of the steam jet in a tangential direction is:

$$v_1 - v_2 = c_o + (c_o - 2u) = 2(c_o - u).$$

The absolute outlet velocity $c_o - 2u$ is positive, as this velocity has an opposite direction to c_o.

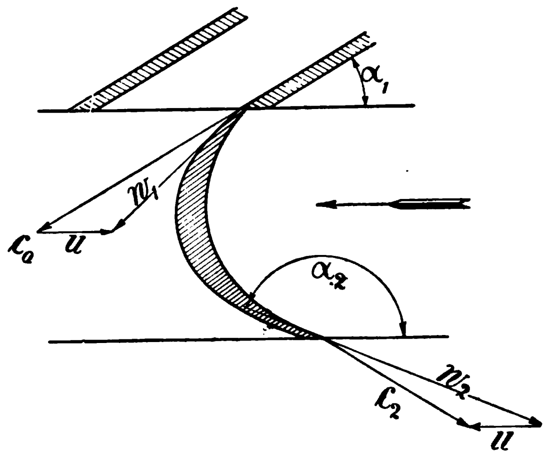

Fig. 11.

The work done by the steam is:—

$$W = \frac{Q}{g} \cdot 2 \cdot (c_o - u)u,$$

and becomes a maximum when $u = \frac{c_o}{2}$, so that:

$$W_{max} = \frac{Q}{g} \cdot \frac{c_o^2}{2},$$

from which the efficiency will be:

$$\eta = \frac{W}{W_{max}} = \frac{2(c_o - u)u}{\frac{c_o^2}{2}} = 4\left[\frac{u}{c_o} - \left(\frac{u}{c_o}\right)^2\right].$$

Fig. 12 shows how the efficiency varies. The values of $\frac{u}{c_o}$ are entered as abscissæ, and the efficiencies as ordinates. At the value of $\frac{u}{c_o} = 0$ and $\frac{u}{c_o} = 1$, the efficiency becomes zero, and is a maximum at $\frac{u}{c_o} = \cdot 5$. In accordance with the character of the formula for $\frac{W}{W_{max}}$, the efficiency curve is a parabola. This investigation shows that:

"*The theoretical maximum efficiency of impulse blading is obtained when the blade velocity is one-half the velocity at which the steam leaves the nozzle.*"

We can make the same deductions for more complicated blade shapes where the direction of the inlet and outlet velocities form acute angles with the direction of the blade velocity. In doing so it must not be overlooked that only the tangential components of the inlet and outlet velocities of the steam are entered into the formula for the work done by the steam.

§ 37. **Ideal Reaction Turbine.**—The reaction turbine in its simplest form (see § 32) consists of a wheel rotating round an axis, and

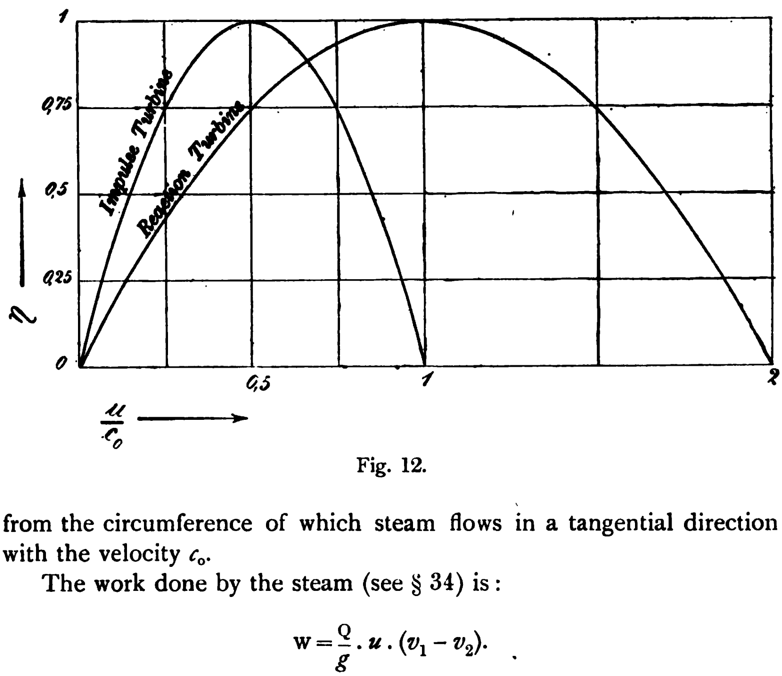

Fig. 12.

from the circumference of which steam flows in a tangential direction with the velocity c_0.

The work done by the steam (see § 34) is:

$$W = \frac{Q}{g} \cdot u \cdot (v_1 - v_2).$$

The change in velocity $v_1 - v_2$ is in this case equal to the outlet velocity c_0, so that the above formula could be written:

$$W = \frac{Q}{g} \cdot u \cdot c_0,$$

if the steam issuing from the wheel had not to be continually accelerated to the velocity u.

To accomplish this a kinetic energy equal to $\frac{Q}{g} \cdot \frac{u^2}{2}$ is necessary, and has to be obtained from the heat energy contained in the steam. Deducting this from the total heat energy there remains as available useful work:

$$W = \frac{Q}{g} \cdot u \cdot c_0 - \frac{Q}{g} \cdot \frac{u^2}{2} = \frac{Q}{g} \cdot u \cdot \left(c_0 - \frac{u}{2}\right).$$

This becomes a maximum for $u = c_0$, and the maximum work therefore is:

$$W_{max} = \frac{Q}{g} \cdot \frac{c_0^2}{2},$$

and the efficiency:

$$\eta = \frac{W}{W_{max}} = \frac{u\left(c_0 - \frac{u}{2}\right)}{\frac{c_0^2}{2}} = 2\frac{u}{c_0} - \left(\frac{u}{c_0}\right)^2.$$

The efficiency curve in this case will also be a parabola, but the efficiency will become a maximum when $\frac{u}{c_0} = 1$, when the blade velocity is equal to the outlet velocity of the steam from the wheel.

A comparison of this result with the efficiency curve of the impulse turbine shows that to obtain the maximum efficiency with a reaction turbine a higher blade velocity is necessary than with an impulse turbine, for in the latter the maximum efficiency is reached when the blade velocity is equal to half the steam velocity, whereas with reaction blading this is only the case when the blade and steam velocity are equal.

These conclusions only hold good for a degree of reaction of 100 per cent., which is never approached in steam turbines and has only been chosen to give an idea of the nature of the reaction.

With reference to the calculation of the efficiency of actual blading we refer to Section IV. below in connection with the calculation of Parsons turbines and to Stodola, "Steam Turbine," 4th German Edition or (3rd) English Edition.

§ 38. **General Remarks on Pressure and Velocity Stages.**—We have seen from the above that to obtain a satisfactory efficiency the blade velocity must have a certain ratio relatively to the steam velocity. By attempting to utilise the whole drop from the boiler to the condenser pressure in one wheel a speed of the wheel would be required which would make it impossible to design them of the existing materials, as they would be unable to withstand the loading due to centrifugal force. Apart, however, from this factor it is in the case of marine steam turbines quite impossible to exceed a certain speed without a low propeller efficiency.

For marine turbines it is therefore essential that the heat drop between the boiler and condenser is divided over a number of groups of blades or stages, so that each stage has only to deal with a heat drop of such a value that a favourable degree of efficiency can be maintained at a given blade speed.

This division of the heat drop can be accomplished as follows:—

1. By arranging for a number of stages, each consisting of one row of guide blades and one row of revolving blades, so that each stage will deal with a certain proportion of the heat drop. The total heat drop is divided into as many single drops as there are stages. The blading can be either of the impulse or the reaction type. An example of the former is the Rateau, and of the latter the Parsons turbine.

2. By dividing the drop over a number of stages consisting, instead of a single set of fixed and revolving blades, of two, three, or four sets of guides and rotating blades, and in which the energy corresponding to a given heat drop is all transformed into velocity at the inlet to each stage. The velocity is then successfully converted into work in the different sets of fixed and rotating blades, each set of rotating blades absorbing part of the velocity. In each pressure stage there is consequently a velocity stage. The pure Curtis turbine used in the U.S. Navy and the Royal Japanese Navy are designed on this principle.

3. By combining the two and constructing the high-pressure part in accordance with 2 above, and the low-pressure part in accordance with 1. The A.E.G. turbine is an example of this design.

§ 39. **Pressure Stages.**—If we assume that equal power is developed by the steam in each stage, and further, if we consider a pure impulse or a pure reaction turbine with n stages, then the power developed by each stage is equal to $\frac{W}{n}$. The steam velocity due to the whole heat drop is equal to (see § 24):

$$c_o = \sqrt{2gW} = 91{\cdot}5\sqrt{h} \text{ metric units,}$$

$$= 223{\cdot}5\sqrt{h} \text{ English units,}$$

and for each stage

$$c_n = \sqrt{2g\frac{W}{n}}.$$

By dividing the formulæ for c_o and c_n we obtain

$$\frac{c_o}{c_n} = \sqrt{n},$$

i.e., the outlet velocity per stage is in inverse proportion to the square root of the number of pressure stages. As the most economical blade velocity is proportional to the outlet velocity of the steam (see § 36), the following holds good:—

On the assumption of equal efficiency the blade velocity of a turbine with n stages can be lower in the proportion of $\sqrt{n}$ to 1 than the blade velocity of a turbine in which the whole heat drop is utilised in a single stage.

In a reaction turbine in which the same quantity of heat is converted into velocity in the guide blades as in the rotating blades (that is, with a reaction coefficient of 50 per cent.), the steam velocity per stage becomes

$$c_n = \sqrt{g\frac{W}{n}},$$

and,

$$\frac{c_o}{c_n} = \sqrt{2n}.$$

Example 1.—Let the diameter of the blade circle of a pure impulse turbine be 2 m. (6·56 ft.), the number of revolutions 700 per minute, the pressure drop from 16 atm. absolute (235 lbs. per sq. in.) to 90 per cent. vacuum, corresponding to an absolute pressure of ·1 atm. (1·47 lbs. per sq. in.), then the heat drop will be 181 cal. (326 B.Th.U.) on the assumption of adiabatic expansion.

At an efficiency ratio equal to 100 per cent. the steam velocity would have to be twice the blade velocity, and

$$c_n = 2 \,.\, 73{\cdot}5 = 147 \text{ m. per sec., or } 482 \text{ ft. per sec.}$$

The theoretical steam velocity for a heat drop of 181 cal. (326 B.Th.U.) will be:

$$c_o = 91{\cdot}5\sqrt{181} = 1{,}235 \text{ m. per sec.}$$

$$c_o = 223{\cdot}5\sqrt{326} = 4{,}030 \text{ ft. per sec.,}$$

and the number of stages,

$$n = \left(\frac{c_o}{c_n}\right)^2 = \left(\frac{1{,}235}{147}\right)^2 = 70 \text{ metric units,}$$

$$n = \left(\frac{c_o}{c_m}\right)^2 = \left(\frac{4{,}030}{482}\right)^2 = 70 \text{ English units.}$$

Example 2.—With a Parsons turbine with the usual reaction coefficient of 50 per cent.,

$$\frac{c_o}{c_n} = \sqrt{2n} = 1{\cdot}41\sqrt{n} \text{ and } n = {\cdot}5\left(\frac{c_o}{c_n}\right)^2,$$

and for the values of example 1,

$$n = \cdot 5\left(\frac{1{,}235}{73\cdot 5}\right)^2 = \text{approximately } 140 \text{ stages metric units,}$$

$$n = \cdot 5\left(\frac{4{,}030}{241}\right)^2 = \quad ,, \quad 140 \quad ,, \quad \text{English units.}$$

§ 40. **Velocity Stages.**—As explained in § 38, 2, the velocity of the steam can be divided over a number of rows of blades, so that

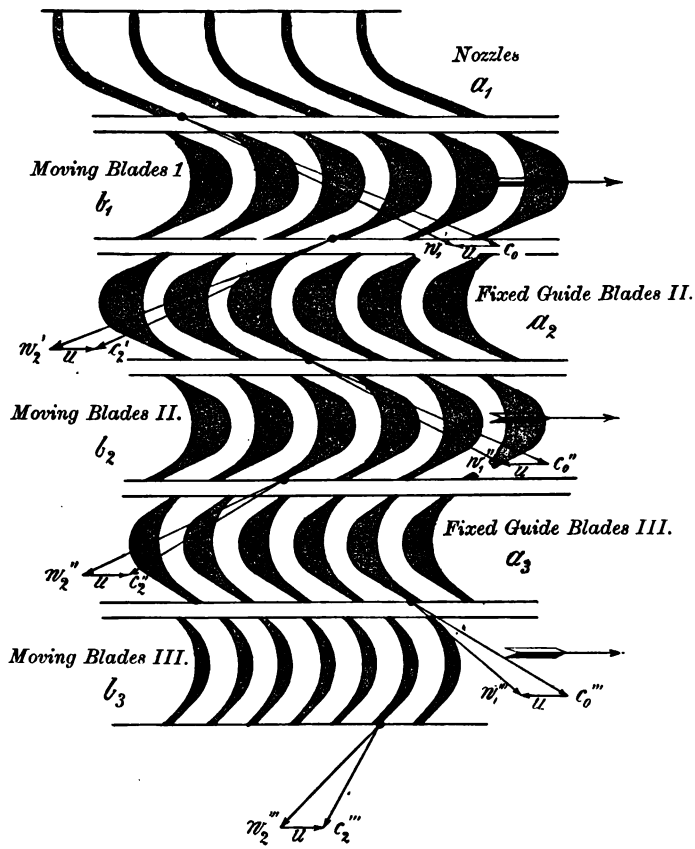

Fig. 13.

each row only converts a part of the steam velocity into kinetic energy. Between two adjacent rows of moving blades stationary blades are required, so as to redirect the steam to the following row of moving blades (see Fig. 13). The velocity of the steam is gradually reduced in passing from the inlet nozzle to the last row of blades, so that in an ideal turbine the absolute outlet velocity from the last row of rotating blades will be zero. Fig. 13 shows a wheel with three velocity stages, the first of which is formed by the nozzle a_1 and the first row of rotating blades b_1, the second stage by the fixed blades a_2 and the second row of rotating blades b_2, the third stage by the fixed blades a_3 and the rotating blades b_3. In the illustrations the vector diagrams of the velocities have been entered, showing how by combining the outlet velocity of the steam, and the blade velocity a gradual reduction of the outlet velocity results.

If in the following all losses are neglected, and the outlet velocity of the steam from the nozzles is denoted by c_o, if, further, the blade velocity u is combined with c_o, the resultant is the relative inlet velocity into the first row of moving blades b_1 is equal to w_1'. The relative outlet velocity w_2' from the first row of moving blades will be equal to the inlet velocity w_1'. The absolute outlet velocity from the rotating blades b_1 is obtained from the velocity diagram as c_2'. The direction of this velocity is changed in the first row of stationary blades, a_2, and again neglecting all losses the outlet velocity from this row of blades is also equal to c_2', but is denoted for the sake of uniformity by c_o''. Following the above line of reasoning the relative inlet velocity of the second row of rotating blades is w_1''.

For the second row of rotating blades the relative outlet velocity is $w_2''=w_1''$. From the vector diagram the absolute outlet velocity from the second row of rotating blades is c_2''.

The direction of the flow of steam is again changed in the second row of stationary blades, so as to impinge without shock on the third row of rotating blades, the outlet velocity from the second row of fixed blades being $c_o'''=c_2''$. The relative inlet velocity into the third row of rotating blades is w_1''', and the relative outlet velocity $w_2'''=w_1'''$. From the vector diagram the absolute outlet velocity from the last row of blades is c_2'''.

How the use of velocity stages influences the number of rows of blades is shown by the following consideration :—

Assuming that the steam jet could be deflected in the blades through an angle of 180°, the outlet velocity from each row of blades would be equal to the inlet velocity less twice the blade velocity. If there are n rows of blades the outlet velocity of the last row would be equal to the inlet velocity into the first row of blades less $n \,.\, 2u$. On the assumption of an ideal efficiency of $\eta=1$, the outlet velocity from the last row of blades will be $=0$, and consequently

$$c_a=c_o-n\,.\,2u=0,$$

$$n=\frac{c_o}{2u} \text{ and } c_o=2u\,.\,n.$$

Let h be the heat drop in a single row of blades,

h_n the heat drop in a wheel containing n rows of blades running at the same blade velocity as the single row of blades.

Then from the above we have the relation—

$$h_n = h \cdot n^2.$$

This leads to the conclusion that by:—

Using a multiple row wheel with n velocity stages at the same blade velocity and efficiency as a wheel with a single row of blades, it is possible to utilise a larger heat drop in the former in the proportion of $n^2 : 1$. This shows further that although the efficiency of multiple row wheels is lower than that of a wheel with a single row of blades, yet the use of the multiple row wheel is a means of reducing both the weight and the space occupied by a turbine. For this reason the multiple row wheel with velocity stages is an important constructive feature. As Curtis was the first to make commercial use of a multiple row wheel with velocity stages, and the Curtis turbines of the present day all consist of such wheels, this combination is usually called a Curtis wheel

§ 41. **Combination of Pressure and Velocity Stages.**—For marine steam turbines, except when they consist of pure pressure stages like the Parsons turbines, practically all modern designers use a combination turbine containing both velocity and pressure stages. The velocity stages generally form the high-pressure part of the turbine, and consist, as a rule, of a series of multiple row wheels.

After passing the blading of the first wheel the velocity of the steam is destroyed, and the steam is then passed through nozzles into the second multiple row wheel (see § 49, Calculation of the A.E.G. Turbine).

§ 42. **General Remarks on Blade Dimensions.**—In dimensioning the blades of marine turbines the following rule should be observed: *The blade sections are to be as small as possible.*

The reasons for this are, first, saving in weight, and secondly, saving in space due to the reduction in the axial length of the turbine.

An extreme reduction of the size of the blades must naturally be avoided for manufacturing reasons. They must be sufficiently strong to enable them to be inserted into the grooves provided for them without difficulty or chance of damage, and to enable the shrouding strip, if such is required, to be riveted on to them. This is particularly necessary for the low-pressure blades, which, in order to obtain the necessary area for the large volume of steam, are frequently of considerable length.

The longest blades of the turbines on the Cunard liners "Mauretania" and "Lusitania" are 559 mm. (22 in.) long. The width of marine turbine blades varies from 10 to 25 mm. (about $\frac{3}{8}$ to 1 in). The maximum thickness of the blades at about the centre of the section depends largely on the size of the inlet and outlet angle. This thickness is usually greatest in the blades of the impulse turbines of the Curtis type, as the space between the blades must be the same throughout the entire passage, but is a minimum in the blade of the low-pressure drum. The pure Curtis turbine has comparatively heavy blade sections, and therefore the advantage that the blading is very strong.

In fixing the size of blade sections theoretical considerations play no part, as the function of the blade is to change the velocity of the steam jet by gradually changing the direction of flow. For this purpose the smallest sections which can be easily manufactured are quite

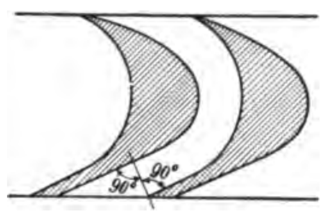

Fig. 14.

suitable. Larger blade dimensions would only result in increased friction losses.

The pitch of the blades is determined by their size, as the blades have to be placed so close to one another that they form a suitable channel and guide for the steam jet. This condition is fulfilled if a vertical line from the inner edge of one blade on to the back of the following blade still cuts that part of the blade at which the tangent of the blade does not vary from the outlet direction (see Fig. 14). Further particulars as to shape and size of the blade are given in Sections III. and IV. below.

§ 43. **Outlet Losses.**—To obtain the maximum efficiency, the steam should leave the last row of blades without any velocity. Practically this is not possible, and cannot even be approximately attained, as the steam passage between the blades of the last rows must be wide on account of the large volume of working fluid, so that it is not possible for the blades at the outlet edge to approach the tangential direction, which is necessary to obtain a minimum outlet velocity.

The outlet losses are equal to the kinetic energy stored in the steam at the absolute outlet velocity c_2, or for a quantity of steam equal to Q per second are:

$$h_v = \frac{AQ}{g} \cdot \frac{c_2^2}{2}.$$

The actual outlet losses in marine turbines are about 2 to 6 cal. per kg. of steam (3·6 to 11 B.Th.U. per lb. of steam) corresponding to 2 to 4 per cent. of the total available heat drop.

§ 44. **Other Losses.**—The losses to be accounted for in a turbine are the following:—

1. Steam friction losses including eddies and impact losses.
2. Clearance losses.
3. Ventilation losses.
4. Mechanical friction losses.

1. The losses resulting from the friction of the drums or discs rotating in steam are so small in marine steam turbines that it is hardly possible to calculate them. The losses due to the fluid friction of the steam in the nozzles, guide, and moving blades is, however, considerable. These losses are allowed for in the calculation by multiplying the steam velocities with coefficients which are less than unity (see Sections III. and IV. on the Calculation of the A.E.G. and Parsons Turbine).

2. The clearance losses include the blade clearance losses, losses in the dummies and glands (see Sections III. and IV. below).

3. In marine steam turbines the ventilation losses are of considerable importance, as nearly all these turbines contain rotating parts which are running light when steaming both ahead and astern. If a row of blades rotates in air at atmospheric pressure, especially in a contrary direction to that for which it is bladed, a considerable resistance has to be overcome, and this happens to be the case with a reversing turbine direct coupled to the ahead turbine, so that the design is always arranged for that section of the turbine which is for the time running light to rotate in vacuum. The absence of a medium tending to act as a break will then considerably reduce the ventilation losses.

These are, however, not the only ventilation losses which occur, as the wheels or drum, especially when partial steam admission is employed, rotate in a space surrounded by steam, and cause ventilation losses which have to be overcome.

It is clear that these losses cannot be calculated with any degree of accuracy, and that the allowance to be made for these must be ascertained experimentally for each type of turbine. Generally, however, the following holds good:—

The work required to overcome the ventilation losses is proportional to the specific weight of the steam as well as to the third power of the number of revolutions (approximately).*

In a complete marine turbine installation the ventilation losses amount to about 2 to 4 per cent. of the total output when steaming ahead, but when steaming astern these losses will be at least doubled, depending on how many wheels or rows of blades there are in the ahead turbine.

4. The bearing friction can be calculated on the usual lines.

If, W is the weight of the rotating parts in kg. (or lbs.),
d „ diameter of the shaft in mm. (ins.),
n „ number of revolutions,
ρ „ friction coefficient,

then the total bearing friction R in horse-power will be:

$$R=\frac{W.\rho.d\pi n}{60.75}=\frac{W.\rho.d.n}{1,430} \text{ metric units.}$$

$$R=\frac{W.\rho.d\pi n}{33,000}=\frac{W.\rho.d.n}{10,500} \text{ English units.}$$

The friction in the thrust-block also requires to be taken into consideration, but if the friction coefficient for the main bearings is taken on the high side, then the allowance made for these will be sufficient to cover the thrustblock friction.

As forced lubrication is always used for the bearings of steam turbines, the co-efficient of friction will be small and will not exceed ·01 to ·005.

By working out a concrete example it will be seen that the bearing friction is of much smaller importance than the ventilation losses.

§ 45. **Indicated Output of a Turbine.**—The indicated output of a turbine is sometimes referred to, denoting the work done by the steam on the buckets of the turbine. The effective output obtained from the turbine spindle is equal to the indicated output, less the losses due to steam and mechanical friction. The leakage losses through the glands, dummy pistons, &c., *i.e.*, all losses resulting from the working medium, are already allowed for in the indicated output. In well-designed turbines, the effective output when steaming ahead is about 5 to 6 per cent. less than the indicated output. This indicated output must not be confused with the indicated horse-power of an equivalent reciprocating engine (see § 5).

Accordingly the indicated efficiency of a steam turbine is: $\eta_i=\frac{N_i}{N}$, N being the work corresponding to the heat drop (see § 18).

* See Stodola, "Steam Turbine," English Ed., p. 129, 4th German Ed., p. 120.

§ 46. **Critical Velocity.**—The critical velocity and the critical number of revolutions are factors which do not require the consideration of the designer of reciprocating engines, but which must be taken into account in the design of turbines with their heavily loaded shafts running at high speeds.

A mass rotating round an axis which does not pass through the centre of gravity of that mass creates a centrifugal force tending to bend the shaft.

If, for instance, the weight of this eccentric mass is $P=10$ kg. rotating at a radius of 1 m. from the centre of the shaft at 700 revs. per min., then the velocity of the mass will be:

$$u=\frac{2r\pi \cdot n}{60}=\frac{2 \cdot 1 \cdot 3{\cdot}14 \cdot 700}{60}=73{\cdot}2 \text{ m.},$$

and the centrifugal force

$$c=\frac{P}{g}\cdot\frac{u^2}{r}=\frac{10}{9{\cdot}81}\cdot\frac{73{\cdot}2^2}{1}=5{,}450 \text{ kg.}$$

From this enormous value of the centrifugal force it is apparent how any eccentric loading of a turbine shaft must be avoided.

Even the small amount of out of balance of the discs and drum which occurs in manufacture is of such a nature that it cannot be neglected, and the turbine shafts must therefore be calculated to run at a speed at which there is no danger of excessive deformation of the shaft. To obtain this turbine shafts are run at a velocity considerably below the so-called critical speed.

The idea of the critical speed may be explained as follows:—

We may assume that the rotating part of a steam turbine such as the shaft, drum, and discs of the total weight P are slightly out of balance, so that the centre of gravity of the whole rotating mass is displaced from the centre of rotating by the amount e, then when rotating, the position is that the mass P will be moving with the small eccentricity equal to e round the centre of the shaft. It is evident that due to the centrifugal force this eccentricity will increase with increasing number of revolutions. This eccentricity increasing with the number of revolutions or the angular velocity ω may be denoted by y. Then the centrifugal force in any moment will be:

$$C=\frac{P}{g}\cdot\frac{u^2}{r},$$

and as the velocity u at any moment is ωy and the radius at any moment y, then:

$$C=\frac{P}{g}\,\omega^2 y.$$

Opposed to the deflection of the shaft caused by this force is the elasticity of the shaft.

If K is the force necessary to deflect the shaft 1 cm., then the force necessary to deflect the shaft by the amount $y-e$ from the centre is $K(y-e)$.

At that instant in which the slightest external force acting on the shaft is able to cause increasing deflection up to breakage of the shaft, the centrifugal force and the stiffness of the shaft are in equilibrium, or,

$$K.(y-e)=\frac{P}{g}\omega^2 y,$$

from which

$$y=\frac{e}{1-\frac{P}{K}.\frac{\omega^2}{g}}$$

If $\frac{P}{K}.\frac{\omega^2}{g}=1$, then the value of y becomes infinite, *i.e.*, the centrifugal force will bend the shaft until it breaks, even if the original eccentricity e is infinitesimal. Let $\omega\infty$ be the angular velocity for this case, then :—

$$\omega\infty=\sqrt{\frac{K.g}{P}}.$$

This shows that even in a perfectly balanced shaft the angular velocity $\omega\infty$ must be avoided, as otherwise the smallest deflection through an external force would be sufficient to cause breakage of the shaft if the shaft were not guided (by bearings for instance) to prevent excessive bending.

The angular velocity $\omega\infty$ and the corresponding number of revolutions $n\infty$ are called the *Critical Angular Velocity* and the *Critical Number of Revolutions.*

As:

$$\frac{2\pi n}{60}=\omega, \text{ or } \omega=\frac{\pi.n}{30},$$

then:

$$n\infty=\omega\infty\frac{30}{\pi},$$

$$n\infty=300\sqrt{\frac{K}{P}}.$$

It is therefore necessary to design the shaft that the critical number of revolutions is above the normal number of revolutions. Usually the value of

$$\frac{n\infty}{n}=2 \text{ to } 3.$$

The greater this value becomes, the greater is the permissible eccentricity of the rotating part without danger of any breakage of the shaft.

§ 47. **Calculation of the Critical Number of Revolutions.**—The calculation in the simple case of a single disc on a shaft is not difficult.

If a shaft of a length of $2a$ (Fig. 15) between bearing centres is weighted in the centre with a disc G, then the force K required to bend the shaft by 1 cm. is:

$$K = \frac{6 . J . E}{a^3}$$

and the critical angular velocity,

$$\omega\infty = 300\sqrt{\frac{6 . J . E}{a^3 G}}.$$

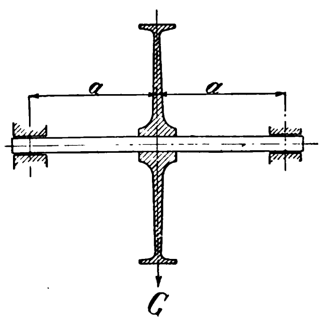

Fig. 15.

If we neglect the weight of the shaft, but assume a constant diameter of the shaft of 100 mm. and $2a = 100$ cm., and $G = 1{,}000$ kg., the shaft being of forged steel, then:

The moment of Inertia $J = 490{\cdot}9$ cm.^{4},
The modulus of Elasticity $E = 2{,}000{,}000$ kg./sq. per cm.,

and,

$$n\infty = 300\sqrt{\frac{6 . 490{\cdot}9 . 2{,}000{,}000}{50^3 . 1{,}000}} = 300\sqrt{47{\cdot}1} = 2{,}060 \text{ revs. per min.}$$

If this shaft had to rotate at 2,000 revs. per min., the factor of safety against breakage would be too low, and the thickness of the shaft would have to be increased. If this was 150 mm., then the moment of inertia would be $J = 2{,}485$, and the critical number of revolutions would be $n\infty = 4{,}630$, or more than twice the normal running speed, which allows of a sufficient margin.

§ 48. **Critical Number of Revolutions of a Multiple Loaded Shaft.**—For calculating this we shall use the fundamental law of Mohr for determining the elastic curve of deflected beams.

In Fig. 16 a shaft is shown carrying a Curtis wheel and two discs supporting a drum. The weight of the disc be $G_1 = 500$ kg., and the weight on the drum discs inclusive of the drum $G_2 = 800$ kg. and $G_3 = 400$ kg.

The shaft is assumed to be of constant diameter of 300 mm. Neglecting the weight of the shaft and assuming an angular velocity of $\omega = 100$, then the deflection of the shaft due to the centrifugal force of the three masses will be approximately as shown in curve $A\,a\,c\,f\,B$.

The deflection at each point will be:—

$$r_1 = ab = 12 \text{ mm.}$$
$$r_2 = ed = 17 \text{ mm.}$$
$$r_3 = fg = 14 \text{ mm.},$$

and the centrifugal force due to these deflections:

$$C_1 = m_1 r_1 \omega_1^2 = \frac{500}{9{\cdot}81} \,.\, {\cdot}012 \,.\, 100^2 = 6{,}120 \text{ kg.}$$

$$C^2 = m_2 r_2 \omega_2^2 = \frac{800}{9{\cdot}81} \,.\, {\cdot}017 \,.\, 100^2 = 13{,}860 \text{ kg.}$$

$$C_3 = m_3 r_3 \omega_3^2 = \frac{400}{9{\cdot}81} \,.\, {\cdot}014 \,.\, 100^2 = 5{,}710 \text{ kg.}$$

From this the bending moment areas A_1, a_1, e_1, f_1, B_1, Fig. 17, can be found. In this diagram 1 mm. = 300 kg., the length of the shaft being one-twentieth full size.*

In the diagram of forces $A = 15{,}000$ kg. is chosen at 50 mm.

To obtain the elastic curve the bending area must be considered as load area, *i.e.*, the ordinates $\overline{a_1 b_1}$, $\overline{c_1 d_1}$, &c., indicate the load on the shaft at the various points. As the beam is to the scale of one-twentieth, 1 sq. mm. is equal to $1 \cdot 20^2 = 400$ sq. mm. = 4 sq. cm.

The load P_1 on the shaft over $A_1\, b_1$ is equal to the area $A_1\, a_1\, b_1$, that of P_2 over $b_1\, d_1$ equal to $b_1\, a_1\, c_1\, d_1$, &c., the point of application being at $S_1\, S_2$, &c. With this load a new bending moment area, Fig. 18, can be obtained, the elastic curve of this showing the required bending of the shaft. In this diagram for the loads $P_1\, P_2$, &c., 1 mm. = 200 sq. cm. of the load area and the forces acting on the centre of gravity $S_1\, S_2$, &c., are:—

$$P_1 = 12{\cdot}4 \text{ mm.}$$
$$P_2 = 40{\cdot}5 \text{ mm.}$$
$$P_3 = 17{\cdot}6 \text{ mm.}$$
$$P_4 = 15{\cdot}2 \text{ mm.}$$

* It must be noted that Figs. 16, 17, and 18 are reduced in proportion of 1 : 2·5.

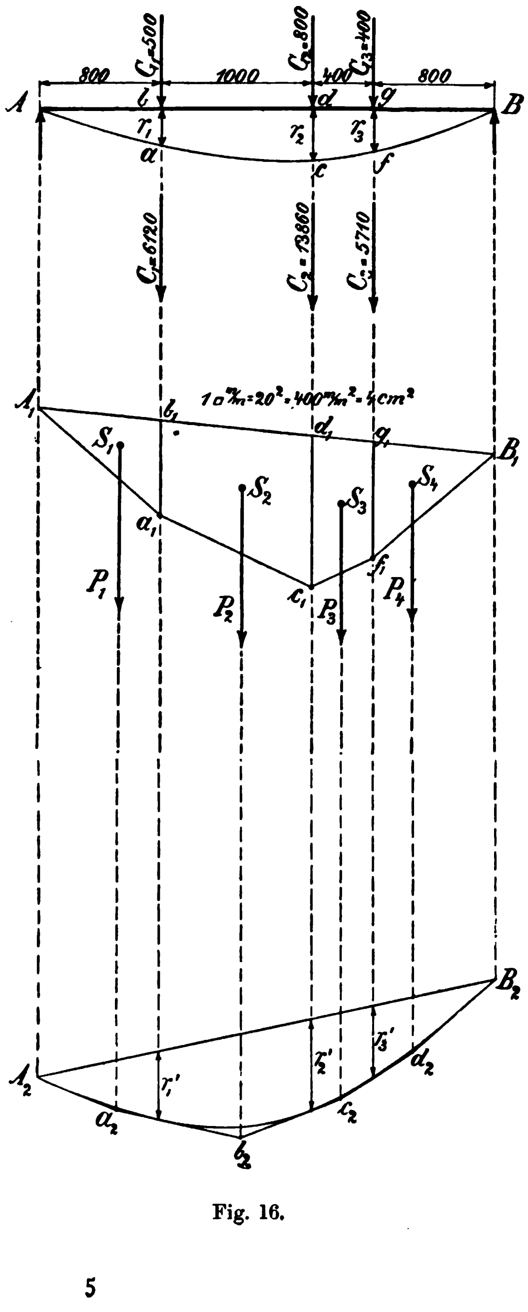

Fig. 16.

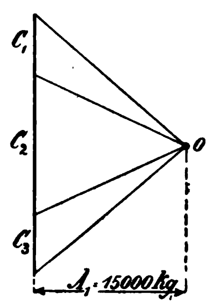

Fig. 17.

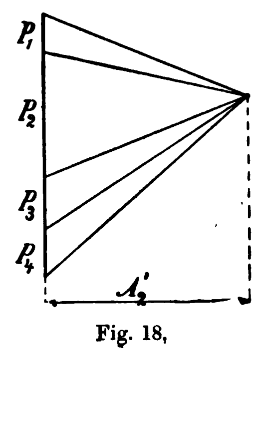

Fig. 18.

The distance A_2 in the diagram of forces, Fig. 18, must, in accordance with Mohr's law, be:

$$A_2 = \frac{E \cdot J}{A_1},$$

E being the modulus of elasticity assumed to be 2,000,000 kg., and
J moment of inertia of the shaft area referred to neutral axis assumed to be $= 39,760 \text{ cm}^4$.

Therefore—

$$A_2 = \frac{2,000,000 \,.\, 39,760}{15,000} = 5,300,000 \text{ sq. cm.},$$

and in accordance with the above scale:

$$A_2' = \frac{5,300,000}{200} = 26,500 \text{ mm.}$$

If we take only 1 : 400 of this, A_2' becomes 66·25 mm. and the deflections are magnified 400 times. As the beam is to a scale of 1 : 20 these deflections are magnified twenty times. With $A_2' = 66{\cdot}25$ mm. the new polygon becomes $A_2\, a_2\, b_2\, c_2\, d_2\, B_2$, the sides of which form the tangents to the elastic curve.

From this the actual deflections are:—

$$r_1' = 1{\cdot}10 \text{ mm.}$$
$$r_2' = 1{\cdot}50 \text{ mm.}$$
$$r_3' = 1{\cdot}15 \text{ mm.}$$

The critical number of revolutions of the shaft is one at which, even with the centre of rotation passing through the centre of gravity, the slightest deflection may cause the centrifugal force to become so great that an increase in the deflection occurs. Under these conditions there exists an indefinite equilibrium so that these deflections may assume any value.

As the deflections are further proportional to the centrifugal forces causing them and as the latter increase as the square of the angular velocity, the critical angular velocity becomes:

$$\omega\infty = \omega\sqrt{\frac{r}{r_1}}.$$

To obtain the deflections approximated above, the angular velocities would have to be:

For $$G_1,\ \omega\infty = 100\sqrt{\frac{12}{1{\cdot}10}} = \sim 330 \text{ or } n\infty = 3,150.$$

$$G_2,\ \omega\infty = 100\sqrt{\frac{17}{1{\cdot}5}} = \sim 336 \text{ or } n\infty = 3,210.$$

$$G_3,\ \omega\infty = 100\sqrt{\frac{14}{1{\cdot}15}} = \sim 349 \text{ or } n\infty = 3,340.$$

If the original deflections were correctly assumed, then the critical velocities would all have been equal.

If a more accurate result is required the calculated elastic curve would have to take the place of the originally assumed one and the calculation repeated.

If the shaft is of varying diameter, and if consequently the moment of inertia varies, then this will also have to be taken into account in the construction of the bending moment area, Fig. 17.

In this case any one of the various sections of the shaft can be taken for the calculation and the ordinates of the area of moments obtained from this section modified in the ratio of the moments of inertia at the various points.

Thus a new area is obtained which is used for the construction of the second elastic curve.

SECTION III.

Calculation of the A.E G. Turbine.

§ 49. **General Remarks.**—The A.E.G. turbine has been evolved out of the Curtis turbine (see § 77). As the former embodies all the elements of the present-day steam turbine, it can serve to illustrate the general method of calculation of a marine steam turbine, as it is only necessary to select from it one part or the other when the problem of calculating a turbine of some other type is approached.

§ 50. **Introduction.**—Before explaining the method of calculation a short description of the turbine may be given. (A detailed description of the various parts is given in § 67 below.) Referring to Fig. 19 it will be seen that the steam at boiler pressure is admitted by a number of nozzle chambers K through the nozzles D_1 to a four-stage Curtis wheel with four rows of buckets (see § 73). After passing this wheel the steam is collected in the chamber in front of the fixed diaphragm Z_1. This diaphragm is fitted with a number of nozzles D_2 of a larger total area than the first row of nozzles D_1. The steam is directed by the nozzles D_2 to the Curtis wheel C_2 containing three or four rows of buckets, and is then collected in the chamber between the diaphragms Z_1 and Z_2, and again directed through the nozzles D_3 to the third wheel C_3 and so on. Each turbine contains at least one and sometimes a larger number of wheels with four rows of buckets and a further number with three rows, followed, but rarely, by wheels with two rows of buckets. The nozzles in the diaphragm Z_1 Z_2, &c., are of successively increasing area so that they occupy an increasing section of the circumference of the diaphragms until the steam admission to the last wheel covers the whole of the circumference of the last diaphragm. From this last wheel the steam is passed to a drum T containing single rows of blades after the manner of the Parsons turbine, but in most cases bladed on the impulse system (§ 36). The drum is fixed by means of discs or cast spiders to the turbine shaft, the front disc being solid. After leaving the drum the steam flows through the exhaust pipe connection to the condenser. The same cylinder contains the astern turbine R consisting of one or more Curtis wheels and a short drum.

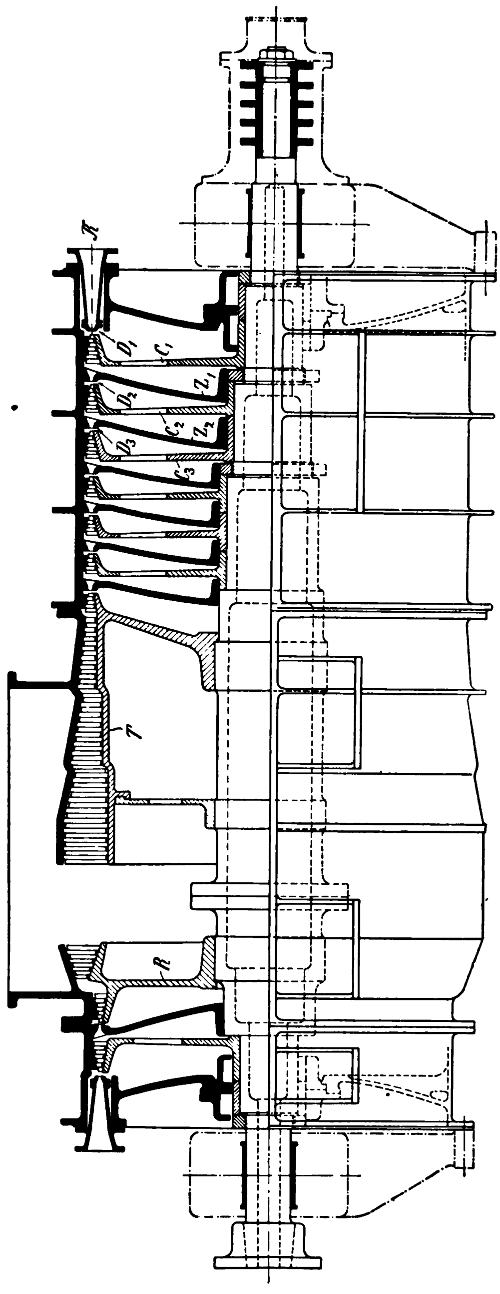

Fig. 19.

§ 51. **The Choice of the Turbine Diameter.**—After the output of the turbine (§§ 4 and 5) and the number of revolutions of the propeller are fixed, the maximum permissible diameter of the turbine cylinder can be fixed. In marine turbines this will depend upon the available space, which, especially in torpedo boats and cruisers, is limited by the lines of the hull, the height available for assembling and dismantling the top cover and rotor, and by the space athwart-ships available for the turbine and condenser. It must not be overlooked that in a plan view the condenser must not be superimposed on the turbine, as this would prevent the raising of the turbine cover.

When dismantling it must be possible after removing the cover to raise the rotor to such an extent that the wheels and diaphragms clear the lower half of the cylinder, and can be withdrawn from the shaft.

The choice of the diameter also depends upon the circumferential speed. Under forced conditions of steaming the blade speed varies between 35 and 60 m. (114 to 197 ft.) per sec. It is hardly possible to select a lower speed than 35 m. (114 ft.), as this would entail a high number of stages and heavy turbines to obtain a reasonable economy. To exceed 60 m. (197 ft.) is not advisable, as the weight of the wheels increase in greater proportion than the diameter, and also because the no-load losses due to the reversing turbine and the ventilation losses of the ahead turbine when steaming astern become considerable, as these losses increase with the 2·7th power of the speed. The increase in weight principally affects the Curtis stages, as the weight of the discs and diaphragms increases with the third power of the diameter.

The higher speeds of 50 to 60 m. (164 to 197 ft.) per sec. have up to the present only been used in torpedo boats, in which utmost economy of space is essential.

In cruisers and battleships it is not advisable to choose speeds below 40 m. (131 ft.) per sec., as the number of stages to obtain a high economy becomes very large, and it is not in the interest of low cost of manufacture to increase the number of these.

For the mercantile marine a greater weight is as a rule permissible, and for turbines for this purpose the speed should be about 50 m. (164 ft.) per sec.

§ 52. **Admission and Exhaust Pressure.**—The admission and exhaust pressure is either given or can be assumed. The following are the admission pressures usually selected:—

For turbines of the mercantile marine - - - - { 16 atm. absolute, about 235 lbs. per sq. in. absolute.

For battleship and cruiser - { 17 to 19 atm. absolute, about 250 to 280 lbs. per sq. in. absolute.

For torpedo boats - - { 19 to 20 atm. absolute, about 280 to 294 lbs. per sq. in. absolute.

The gauge pressures should never be given, as in the calculations only absolute pressures are required, and the use of the former is likely to cause errors and misunderstandings. Low steam pressures (about 10 atm., or 147 lbs. per sq. in.) and a slight degree of superheat have the advantage that the boiler and steam pipe line can be designed lighter than at the high working pressures given above, allowing for the same heat drop over the turbine as with saturated steam at a higher steam pressure. Basing the calculation on the same heat drop a higher turbine efficiency can be obtained with low-pressure superheated steam than with saturated steam at a higher pressure. (With reference to this, see further, §§ 22 and 23.) It will be the aim of the designer to choose a high vacuum, and this should in no case be less than 90 per cent. Vacua of 94 to 95 per cent. can in the present state of the art be considered as the upper limit. As the area of the blade passages at the exhaust end and the size of the exhaust pipe line is limited, it is not always possible to utilise the available heat drop due to the high vacuum. Therefore it does not pay in most cases to instal condensers and pumps of the increased size and weight necessary to obtain a higher vacuum than 95 per cent. (See Section VI.)

§ 53. **Determination of the Rotor Thrust.**—The thrust on the low-pressure turbine drum can be calculated from the rule that the pressure due to the steam on the drum and packing glands minus the propeller thrust shall be slightly less than the propeller thrust. Under forced conditions of steaming, the total thrust due to the steam is assumed to be 180 per cent. of the propeller thrust, so that the resulting pressure on the thrust block is 80 per cent. of propeller thrust and directed astern. The shaft between the turbine drum and thrust block is under tension. An accurate calculation of the variation of the thrust pressure with varying speed of the vessel shows that with decreasing speed the thrust pressure decreases at a quicker rate than the propeller thrust. The thrust pressure becomes zero, and at the lower speeds acts in the same direction as the propeller thrust, although this pressure is only small.

If η_P is the propeller efficiency assumed to be, say, 60 per cent.,
v the speed of the vessel in knots,
P the propeller thrust,
N_s the S.H.P.,

then,
$$P = \frac{N_s \cdot 75 \cdot \eta_P}{\cdot 515 \cdot v}.$$

Fig. 20 shows an example of the loading of the thrust block for a torpedo boat steaming ahead at speeds varying from 10 to 30 knots. The propeller thrust curve is based on experimental tank tests, the pressure curve of the steam from calculations based on tests made at the works of the A.E.G., Berlin. It must be noted that the steam pressure acts on the front end of the drum in an astern direction, *i.e.*, is negative, whereas the thrust of the propeller is directed ahead and is positive.

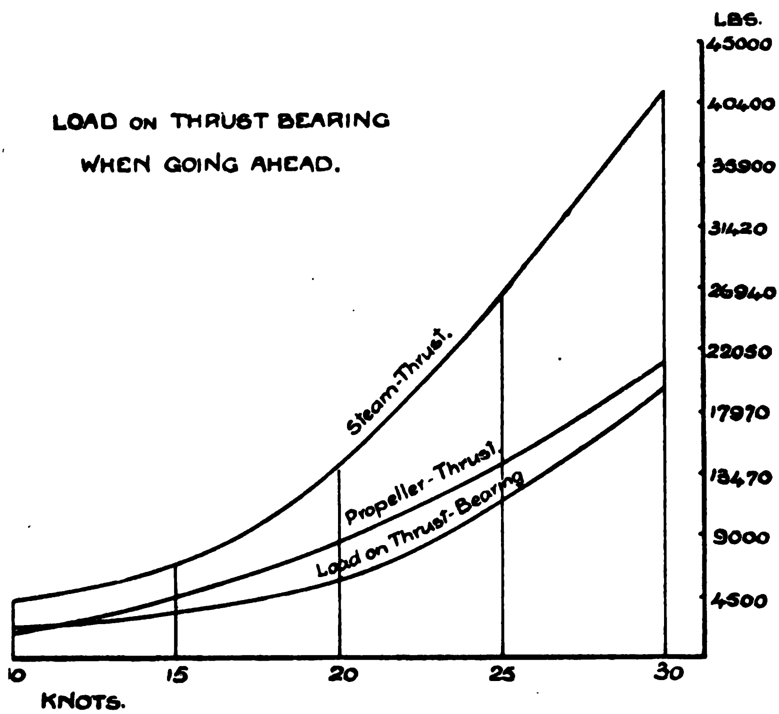

Fig. 20.

In Fig. 20 the pressure on the thrust block at any speed is equal to the length of the ordinate between the curve of the pressure due to the steam and the curve of the propeller thrust. The curve of the thrust block pressure is the loci of these differences. According to this the pressure due to steam at 30 knots is approximately 40,000 lbs., and the propeller thrust approximately $+20{,}700$ lbs., so that the pressure on the thrust block is $-40{,}000+20{,}700=-19{,}300$ lbs., *i.e.*, directed astern and about 93 per cent. of the propeller thrust. The curve shows that above 30 knots the increase in the thrust block pressure is about proportional to the increase in the propeller thrust.

The curve shows further that the pressure on the thrust block rapidly drops with decreasing speed, so that at 20 knots it is only about − 6,200 lbs., or about 74 per cent. of the propeller thrust. Below 20 and down to 10 knots the reduction in the thrust block pressure is small. At 15 knots it is equal, and below 15 knots larger than the propeller thrust, but as the absolute value as well as the speed is small, this is immaterial.

It is also possible that the pressure on the thrust block may pass through O and become negative at a medium speed.

For the purpose of a preliminary calculation the mean nozzle circle diameter (N C diameter) of the first drum stage can be assumed to be equal to that of the Curtis stage. On the assumption that the pressure on the drum is determined as stated above, the blade lengths of the first rows of blade on the drum will vary from 50 to 100 mm. (2 to 4 in.). From this it is possible to approximate the front diameter of the drum. The difference in pressure before and behind the drum is equal to the pressure exerted by the steam, divided by the area of the front drum, and this difference in pressure added to the absolute back pressure at the turbine exhaust will give the pressure at the front end of the drum.

§ 54. **Steam Admission Pressure.**—The steam pressure at the nozzle inlet is about 1·5 to 2 kg. per sq. cm. (21 to 28 lbs. per sq. in.) below the boiler pressure, this allowance being made to compensate for the pressure drop between the boiler and turbine. In the design sufficient space should be provided to enable additional nozzles to be easily added in case the nozzle area should prove too small on test. It should be kept in mind that the steam pressure in all stages rises proportionally to the nozzle area.

§ 55. **Distribution and Calculation of the Heat Drop over the C Stages.**—As soon as the steam admission pressure to the nozzles and to the drum is known, the available adiabatic heat drop for the Curtis stages can be determined from the Mollier diagram. It is assumed that the degree of superheat or the dryness factor when using saturated steam is known or assumed. The latter is generally taken at ·97 to ·98.

On account of the high boiler pressure the first stage is usually a wheel with four rows of blades, as the heat drop over this stage is too great for the use of a three-row wheel. The heat drop h of the first stage is restricted by the permissible steam pressure which should not exceed 8 atm. absolute (117 lbs. per sq. in.) in light and 10 atm. absolute (147 lbs. per sq. in.) in heavier designs, not only on account of the weight of the machinery, but also on account of the difficulty in maintaining the glands steam tight at higher pressures.

The useful heat drop is

$$h_i = h \cdot \eta_i,$$

where η_i is the stage efficiency, *i.e.*, the useful work transmitted to the blades divided by the theoretical work. The value of the efficiency figure is obtained from the efficiency curves which have to be determined experimentally by the manufacturers.

Fig. 21 gives an efficiency curve for a four-row, Fig. 22 for a three-row, and Fig. 23 for a two-row wheel.

The efficiency in these curves is shown as a function of $\frac{c_o}{u}$, where

$$c_o = 91{\cdot}51 \sqrt{h} \text{ metric units.}$$

$$c_o = 223{\cdot}5 \sqrt{h} \text{ English units,}$$

is the theoretical steam velocity due to the adiabatic heat drop over the stage and u the blade velocity at the mean nozzle circle diameter (N C diameter).

The remaining heat drop between the steam condition at the outlet of the first stages and the admission to the drum is equally divided over a number of stages, each consisting of a set of nozzles and wheels with three rows of blades, the number of stages being chosen so that the value of $\frac{c_o}{u}$ approaches that corresponding to the maximum efficiency.

To obtain the required steam admission pressure at the drum it is usual to add about 4 per cent. to the total available heat drop to allow for re-evaporation over the three-row C stages.

The maximum efficiency at the wheel circumference lies between $\frac{c_o}{u} = 5$ to 6, but can only be obtained at the higher blade velocities, as otherwise the number of stages and the heat drop per stage becomes too small, and the test results obtained with three-row wheels absorbing less than 10 cal. (18 B.Th.U.) are too doubtful to enable any definite efficiency values to be given.

The curves opposite (Figs. 21 to 23) are for dry saturated steam ($x = 1$) and on the assumption that the blading is designed for the maximum number of revolutions, *i.e.*, for the smallest value of $\frac{c_o}{u}$ at which the efficiency is required. With falling turbine speed $\frac{c_o}{u}$ increases,

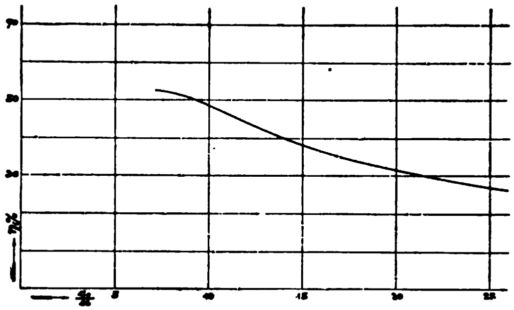

Fig. 21.

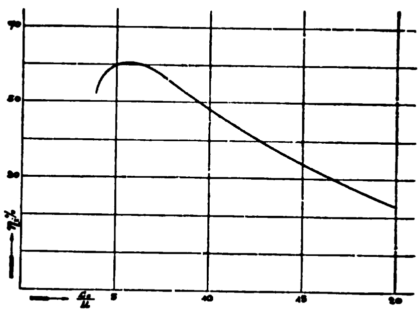

Fig. 22.

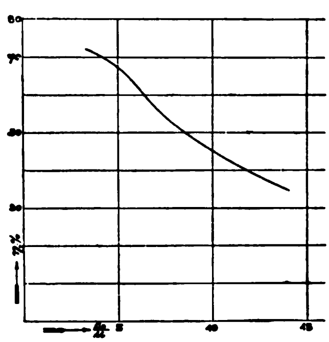

Fig. 23.

and if the efficiency has to be determined for a turbine at cruising speed it is sufficiently accurate to use these curves, although the blading of a cruising stage designed for a large $\frac{c_0}{u}$ will differ considerably from the blading designed for a full speed stage $\left(\text{small } \frac{c_0}{u}\right)$.

Blading designed for a large $\frac{c_0}{u}$ will, at a smaller $\frac{c_0}{u}$, give a lower efficiency as considerable steam impacts occur on the back of the blades. With blading designed for a small $\frac{c_0}{u}$ when running at cruising speed with higher $\frac{c_0}{u}$ the impact occurs on the driving side of the blades, but this has not such a detrimental effect on the efficiency as in the first case.

If the steam at the nozzle inlet is wet, *i.e.*, $x < 1$, a lower output of the turbine will result.

1. Because the available heat drop per unit weight of working fluid is reduced, and
2. Because the efficiency of the nozzles and blading is reduced due to the increased fluid friction.

For approximate calculations it is sufficiently accurate to use the values of the heat drop from the Mollier diagram and the above efficiency curves, making a suitable allowance for the reduction of the efficiency by adding an amount equal to the wetness of the steam to the steam consumption figure.

If the steam is superheated the consumption will be reduced due to :—

1. The increase in the available heat drop, and
2. The reduced friction losses.

The steam consumption has frequently to be guaranteed both with dry saturated as well as with superheated steam. In such a case the turbine is designed for dry saturated steam with the aid of the above efficiency curves and the improvement in the steam consumption with superheated steam is based on test results of similar machines.

§ 56. **Gland Losses.**—Before calculating the output of the C stages the outer and inner gland or stuffing box losses must be fixed so as to arrive at the correct quantity of steam required.

The difference between the total quantity of steam taken by the turbine and the steam taken by the gland will be the steam available for the indicated output of the turbine, and before calculating the C stages the losses in both the outer and inner glands will have to be determined.

§ 57. **Losses through the Outer Glands.**—The outer glands are usually constructed in such a manner that the packing surfaces touch the shaft so that theoretically no loss of steam occurs.

Up to the present, however, there is no type of stuffing box which actually fulfils these conditions, and a certain amount of steam leakage always takes place. It is usually assumed that this leakage is equal to the amount of steam which will pass through an annulus of a given length and having a clearance of ·1 to ·3 mm (·004 to ·012 in.) at the diameter of the shaft.

If the pressure in the first stage is above 1·8 atm. (26·5 lbs. per sq. in.), the quantity of steam passing through the gland will be equal to the critical quantity of steam, assuming the admission pressure to be equal to the pressure in the stage.

This quantity of steam (see § 28) is equal to

$$Q = \cdot 72 A \sqrt{\frac{p_1}{v_1}} \text{ in kg. per hour, or}$$

$$Q = 14{,}370 A \sqrt{\frac{p_1}{v_1}} \text{ in lbs. per hour,}$$

where

p_1 is the absolute pressure of the steam (kg. per sq. cm., or lbs. per sq. in.) in the first stage,

v_1 is the corresponding volume (cb. m. per kg., or cb. ft. per lb.),

A the area of the leakage annulus in sq. mm. or sq. in.

It is usual to work out a set of curves giving the quantity of leakage steam per unit of area.

The friction losses of the steam in passing through the clearance space are rarely taken into consideration on account of the uncertainty of the actual amount of clearance in the gland, and consequently it is possible for the clearance to increase by 20 to 40 per cent. before the losses estimated from the above formula are actually reached.

The whole of the steam admitted to the turbine is utilised in the first stage. The steam passing through the external gland of the first stage is deducted from the steam flowing through the second and following stages until the stage is reached at which the steam from this gland is returned to the turbine.

The high-pressure gland is usually provided with two drain connections, one of which is connected to one of the lower C stages, where the pressure is slightly below the critical pressure of the first stage, the second being connected to a point of the drum where the steam pressure at full load is equal to about ·5 atm. absolute (7·5 lbs. per sq. in.).

Beyond the stage at which the leakage steam from the gland is readmitted, only 50 per cent. of the total leakage is deducted from the total quantity of steam flowing through the turbine. Beyond the second point of admission, *i.e.*, in the drum, the total leakage steam is again utilised (see next paragraph).

§ 58. **Losses in the Intermediate Glands.**—The construction of these is shown in Fig. 38, page 102. They are designed with sufficient radial clearance to prevent contact of any part of the gland with the shaft, so as to avoid any attendance to the gland during running.

In turbines in which the high and low pressure stages are contained in a single casing and in which the last Curtis stage is placed approximately in the centre of the casing, the clearance increases from stage to stage in accordance with the increasing deflection of the shaft. Starting with ·2 to ·4 mm. (·008 to ·016 in.) radial clearance at the first stage, this is increased to about ·6 to 1·0 mm. (·024 to ·04 in.) in the last stage.

If the turbine is designed as a two-cylinder machine in which the first cylinder embraces Curtis stages only, the radial clearances at the first stages will be the same as for the single-cylinder machine, but the maximum clearance at the centre will only be 0·4 to 0·8 mm. (·016 to ·032 in.), this clearance being maintained to the last stage. As the specific volume of the steam is large in the last stages, there is no reason for decreasing the clearances towards the outlet.

The losses in the intermediate glands can be calculated from the amount of the clearances and the critical quantity of steam when the ratio of pressures $p_1 : p_2$ at either side of the gland is above the critical value, or when this is not the case from the formula:

$$Aw = Q \cdot v,$$

in which

w is the theoretical steam velocity calculated from the drop in the nozzle of the corresponding stage, and

v is the specific volume relatively to the back pressure of the gland obtained from the steam tables.

§ 59. **Output of the Curtis Stages.**—The indicated output* of each stage of the turbine is:

$$N_1 = \frac{Q_n h_i}{632{\cdot}3} \text{ Ch. à V. in metric units, or}$$

$$N_1 = \frac{Q_n \cdot h_i}{2{,}545} \text{ H.P. in English units,}$$

where

N_i is the indicated output,

Q_n is the steam passed through the stage, *i.e.*, the total quantity of steam less the amount lost in the glands,

h_i heat drop over the stage.

---

* In steam turbines indicated output denotes the output due to the work of the steam on the blades. To obtain the effective output the ventilation and friction losses must be deducted.

To obtain an approximation a comparison of the gland losses with the steam actually employed in doing work is useful.

§ 60. **Calculation of the Low-Pressure Drum — General Remarks.**—To obtain a minimum number of stages and a maximum radial clearance the greater part of the low-pressure turbine blading consists of single-row impulse sections. Towards the exhaust end, where the amount of clearance is not of great importance, the last rows of blades are designed with a degree of reaction of 50 per cent. For constructional reasons the fixed and rotating blades of the impulse sections are of the same length, so that a small amount of reaction also occurs in these sections.

§ 61. **Heat Drop over the Low-Pressure Drum.**—To the total adiabatic drop over the drum calculated from the steam conditions at the outlet from the last C stages an addition of about 3 per cent. should be made to allow for re-evaporation.

The admission pressure to the drum is usually about 1·2 to 2 atm. absolute (17·5 to 29·5 lbs. per sq. in. absolute).

From the calculation of the Parsons turbine (see § 64), it will be seen that the heat drop cannot be divided equally over the separate stages. Referring to Fig. 29, an available heat drop of 100 + 3 per cent. = 103 cal. (or 180 B.Th.U. + 3 per cent. = 185·5 B.Th.U.) has been assumed, and the heat drop over the stages so chosen that the first row of blades of the drum utilise a smaller drop than the middle stages, as this allows of an increased blade length and lower clearance losses for the first rows of blades. The resulting surplus drop is utilised over the last rows of blades in which the steam velocity has to be increased on account of the limit of blade height being reached.

The curve of total drop in Fig. 29 gives at each point the sum of the separate heat drops utilised up to that point.

For rough estimates it is usually quite sufficient to calculate the blade length of the first and last stages. The dimensions of the intermediate blading can, after a little practice, be determined with the help of efficiency curves calculated in accordance with § 35.

§ 62. **Number of Steps of the Drum.**—From turbines in actual operation the value of $\frac{c_0}{u}$ lies between 3 and 3·5, being calculated from the average heat drop per stage, and from the mean blade velocity. If a large weight is available for the turbine, and if, consequently, a large number of steps can be used, a lower value than 3 can be obtained.

In designing a new turbine, a value of $\frac{c_0}{u}$ is assumed, and the mean drop per stage calculated. By dividing the total heat drop by the mean drop per stage, the number of stages can be obtained.

On account of manufacturing difficulties, it is usual to make a limited number of cylindrical steps, as the continuous increase in blade height corresponding to the increasing steam volume requires a large number of tools and gauges, and thereby considerably increases the cost of manufacture.

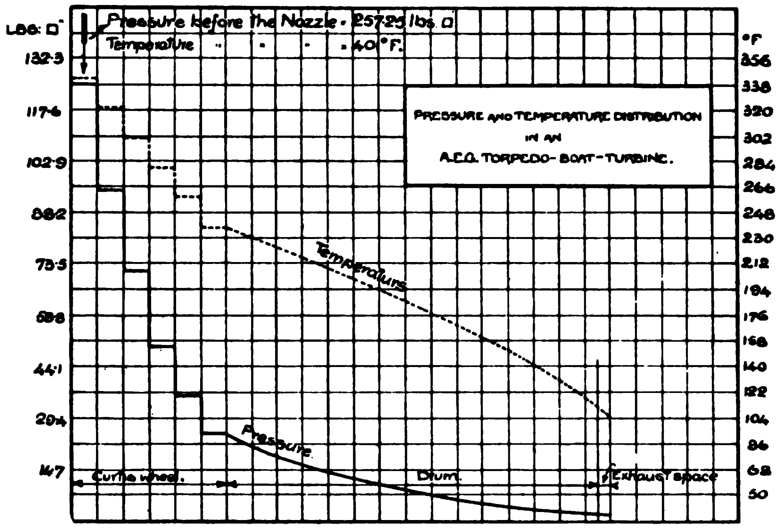

FIG. 24.

Corresponding to the increase in the steam volume, the number of rows consisting of blades of equal height which are assembled together to form a single step decrease towards the exhaust end of the drum. With a drum having thirty stages, a good approximation to the steam volume curve will be obtained if the beginning of the drum is made with steps consisting of not more than five or six rows, and for drums with fifty to sixty stages about twenty rows of blades of the same height.

The height of the blade should increase in proportion to the volume of steam, although this cannot be attained towards the exhaust end of turbines exhausting to a high vacuum, as this would result in impossible blade dimensions. For this reason it is necessary to increase the steam velocity, and consequently the heat drop, over the last group of blades, and to increase the opening of the blades. The higher efficiency at the first rows of blades of the drum, as compared with the efficiency of the blading at the exhaust ends, is partly compensated for by the reduced clearance losses of the last row, which in the first stage may amount to as much as 15 per cent. of the total steam.

§ 63. **Pressures Temperature Distribution.**—Fig. 24 shows the distribution of pressure and temperature over an A.E.G. turbine.

SECTION IV.

Calculation of the Parsons Turbine.

§ 64. **Introduction.**—As previously explained the Parsons turbine is a reaction turbine with a degree of reaction of 50 per cent., consisting of a large number of rows of guide and rotating blades with full steam admission. The calculation of the turbine can be demonstrated by the calculation of a single group of guide blades and rotating blades of constant blade height as in the following example.

§ 65. Calculation of Part of a Parsons Turbine.

Let Q be the steam consumption of the turbine per hour equal to 72,000 kg. per hr. or 159,000 lbs. per hr., or 20 kg. per sec. or 44·2 lbs. per sec.

Assuming the section to be calculated to be the first part of the low-pressure turbine, the steam before entering the first row of guide blades can be assumed to be at rest. Let the steam pressure at this point be 2 atm. absolute (29·4 lbs.), and the dryness factor 94 per cent. The section to be calculated consists of three rows of blades. Let the adiabatic heat drop over the section based upon an assumed drop per expansion (see § 66) be $H = 8$ cal. (14·4 B.Th.U.), corresponding to the final pressure of 1·62 atm. or 23·8 lbs. per sq. in. The mean blade diameter is assumed to be $D = 2{\cdot}4$ m. (7·87 ft.) and the number of revolutions $n = 500$, from which the peripheral speed is:—

$$u = 62{\cdot}8 \text{ m. per sec., or } 206 \text{ ft. per sec.}$$

Let

The heat drop over the 1st guide blade be h_a', 1st moving blade be h_b' $\left.\right\}$ $h_a' + h_b' = h'$

2nd guide blade be h_a'', 2nd moving blade be h_b'' $\left.\right\}$ $h_a'' + h_b'' = h''$

3rd guide blade be h_a''', 3rd moving blade be h_b''' $\left.\right\}$ $h_a''' + h_b''' = h'''$

Further, let—

c_o'	be the theoretical outlet velocity from the 1st guide blade						
c_1'	,,	actual	,,	,,	1st	,,	
		$=\phi c_o'$					
c_o''	,,	theoretical	,,	,,	2nd	,,	
c_1''	,,	actual	,,	,,	2nd	,,	
		$=\phi c_o''$					
c_o'''	,,	theoretical	,,	,,	3rd	,,	
c_1'''	,,	actual	,,	,,	3rd	,,	
		$=\phi c_o'''$					
w'	,,	relative inlet velocity to the 1st moving blades					
w''	,,	,,	,,	,,	2nd	,,	
w'''	,,	,,	,,	,,	3rd	,,	
w_o'	,,	theoretical relative outlet velocity from the 1st moving blades					
w_1'	,,	actual	,,	,,	,,	1st	,,
		$=\psi w_o'$					
w_o''	,,	theoretical	,,	,,	,,	2nd	,,
w_1'''	,,	actual	,,	,,	,,	2nd	,,
		$=\psi w_o''$					
w_o'''	,,	theoretical	,,	,,	,,	3rd	,,
w_1'''	,,	actual	,,	,,	,,	3rd	,,
		$=\psi w_o'''$					
c_2'	,,	absolute outlet velocity from the 1st moving blades					
c_2''	,,	,,	,,	,,	2nd	,,	
c_2'''	,,	,,	,,	,,	3rd	,,	

The coefficients $\phi = \psi$, the values of which are assumed to be ·9, vary according to the type of turbine, and are figures determined solely by the experience of the makers.

As the steam before the first stage is assumed to be at rest, the velocity at the outlet of the first row of fixed blades can be calculated from formula 1:

$$h_a' = A\frac{c_o'^2}{2g} \qquad (1)$$

The heat drop h_a' is an unknown quantity, as only the total heat drop over the whole section has been assumed. If the total heat drop is H, then the drop over each set of blades of the three stages, consisting each of a row of fixed and a row of moving blades, will be $\frac{H}{3} = 2·67$ cal., or 4·8 B.Th.U., on the assumption that the drop is equally divided over the three stages. Experiments have shown that the first stages into which the steam enters from a state of rest consumes a larger drop than the following ones. Against this the velocity increases from stage to stage on account of the increasing specific volume of the steam, and consequently the drop consumed

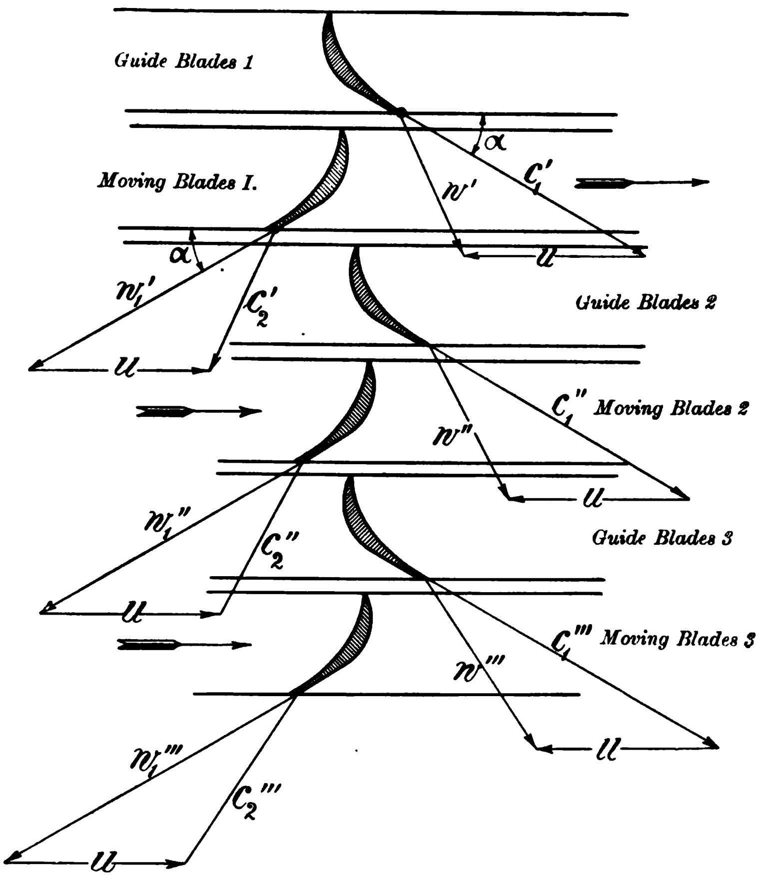

Fig. 25.

per stage will increase. As a first approximation the heat drop over the three stages may be divided as follows:—

$h' = 2{\cdot}5$ cal., or $4{\cdot}5$ B.Th.U.
$h'' = 2{\cdot}5''$ „ $4{\cdot}5$ „
$h''' = 3{\cdot}0$ „ $5{\cdot}4$ „

The reason why h' has been taken to equal h'' is the fact mentioned above of the steam being at rest before the first stage.

Therefore, $h' = h_a' + h_b' = 2{\cdot}5$ cal., or $4{\cdot}5$ B.Th.U.

In formula (1), both h_a' and c_o' are unknown, and to obtain their value a further relationship must be considered. To do so the formula for the outlet of the steam from the first row of moving blades can be used. The following considerations will enable this to be done:—

The kinetic energy of the steam at the outlet of the moving blades is equal to the heat drop consumed plus the kinetic energy at the inlet to the moving blades.

In both cases the relative velocities must be considered.

Taking formula (2) we have:

$$A\frac{w_o'^2}{2g} = h_b' + A\frac{w'^2}{2g}, \text{ or } h_b' = A\frac{w_o'^2}{2g} - A\frac{w'^2}{2g} \qquad (2)$$

It is further assumed that the theoretical outlet velocity from the guide blades is equal to the relative outlet velocities from the moving blades, or:

$$c_o' = w_o',$$
$$c_o'' = w_o'',$$
$$c_o''' = w_o'''.$$

Adding formulæ (1) and (2) we obtain:

$$h_a' + h_b' = h' = 2A\frac{c_o'^2}{2g} - A\frac{w'^2}{2g} \qquad (3)$$

To obtain a solution of this formula it is necessary to consider the

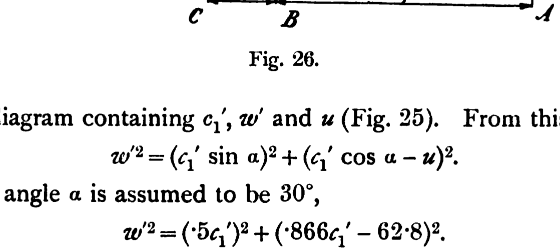

Fig. 26.

velocity diagram containing c_1', w' and u (Fig. 25). From this diagram:

$$w'^2 = (c_1' \sin \alpha)^2 + (c_1' \cos \alpha - u)^2.$$

If the angle α is assumed to be 30°,

$$w'^2 = (\cdot 5c_1')^2 + (\cdot 866c_1' - 62\cdot 8)^2.$$

Inserting this value of w'^2 in (3) we can obtain c_o', which is the only quantity which remains unknown, as

$$c_o' = \frac{c_1'}{\phi}.$$

7

On inserting the actual values in the formula we obtain:

$$c_1' = 98 \text{ m. per sec., and } c_0' = \frac{98}{\cdot 9} = 109 \text{ m. per sec.,}$$

$$c_1' = 321{\cdot}5 \text{ ft. per sec., and } c_0' = \frac{321{\cdot}5}{\cdot 9} = 358 \text{ ft. per sec.}$$

From formulæ (2) and (3) the other unknown velocities are obtained: $w' = c_2' = 53{\cdot}8$ m. ($176{\cdot}5$ ft.), and $w_0' = 59{\cdot}8$ m. per sec. ($196{\cdot}2$ ft. per sec.), and also the values of h_a' and h_b' as follows:

$$h_a' = \frac{A}{2g} \cdot c_0'^2 = \frac{109^2}{8{,}350} = 1{\cdot}42 \text{ cal. metric units,}$$

$$\frac{358^2}{50{,}000} = 2{\cdot}56 \text{ B.Th.U. English units.}$$

The values of the inlet and outlet velocities being known, the velocity diagram can be constructed and the efficiency of the first stage calculated from the formula given in § 35. From Fig. 26 a value of 85 m., or 278·8 ft., for the inlet velocity, and 23 m., or 75·5 ft., for the outlet velocity is obtained. The drop actually consumed, therefore, will be:

$$h_i' = \frac{Au}{g}(\text{AB} + \text{BC}) = \frac{1}{4{,}180} \cdot 62{\cdot}8(85 + 23) = 1{\cdot}62 \text{ cal. metric units,}$$

$$h_i' = \frac{1}{25{,}000}\ 206\ (278{\cdot}8 + 75{\cdot}5) = 2{\cdot}92 \text{ B.Th.U. English units,}$$

and the efficiency:

$$\eta_1 = \frac{h_i'}{h'} = \frac{1{\cdot}62}{2{\cdot}5} = 64{\cdot}8 \text{ per cent. metric units,}$$

$$\eta_1 = \frac{h_i'}{h_1'} = \frac{2{\cdot}92}{4{\cdot}5} = 64{\cdot}8 \text{ per cent. English units,}$$

and the output of this stage:

$$N_i = \frac{h_i}{632{\cdot}3} \cdot 72{,}000 = 185 \text{ Ch. à Vap. metric units,}$$

$$N_i = \frac{h_i}{2{,}545} \cdot 159{,}000 = 182{\cdot}5 \text{ H.P. English units.}$$

Calculation of the Second Stage.

The calculation of the second stage differs from the first in so far as the steam has already an initial velocity on entering the second stage. This velocity is the absolute outlet velocity c_2' from the first row of moving blades. The heat drop in the second row of guide blades is:

$$h_a'' = A\frac{c_o''^2}{2g} - A\frac{c_2'^2}{2g}.$$

The drop consumed in the second row of moving blades is:

$$h_b'' = A\frac{c_o''^2}{2g} - A\frac{w''^2}{2g}.$$

In this formula the assumption is again made that the absolute outlet velocity c_o'' from the second guide blade is equal to the relative outlet velocity w_o'' from the second moving blade.

From the two formulæ for h_a'' and h_b'':

$$h_a'' + h_b'' = h'' = 2 \,.\, A\frac{c_o''^2}{2g} - A\frac{c_2'^2}{2g} - A\frac{w''^2}{2g}.$$

The solution of this formula is as follows:—

The velocity c_2' is known from the velocity diagram of the first stages, and is the absolute outlet velocity from the first moving blade. The relation between c_o'' and w'' is shown in the diagram, so that w'' can be determined from c_o'', or $c_1'' = \phi \,.\, c_o''$ as soon as the angle of outlet is known.

As $h'' = 2{\cdot}5$ cal., or $4{\cdot}5$ B.Th.U. is known, the value of c_o'' can be obtained from the formula for h'', and from the formulæ of the values of h_a'' and h_b'' the individual drop.

After the velocities have been calculated the indicated output and efficiency can be calculated from the components c_1'' and c_2'' of the velocity diagram in the same manner as for the first stages.

The calculations give the following values:—

$c_1'' = 105$ m., or 344 ft.

$c_o'' = \frac{105}{\cdot 9} = 116{\cdot}8$ m., or $\frac{344}{\cdot 9} = 383$ ft.

$w'' = 59{\cdot}4$ m., or $194{\cdot}7$ ft.

$h_a'' = 1{\cdot}28$ cal., or $2{\cdot}3$ B.Th.U.

$h_b'' = 1{\cdot}22$ cal., or $2{\cdot}195$ B.Th.U.

$h_i'' = 1{\cdot}792$ cal., or $3{\cdot}224$ B.Th.U.

$\eta_i = 72$ per cent.

$N_i = 204$ Ch. à. Vap., or 201 H.P.

THE CALCULATION OF THE THIRD STAGE

can be carried out in the same manner as of the second stage.

The results of the calculation for the above example are:—

h_a''' heat drop in the guide blades - - $1{\cdot}57$ cal., or $2{\cdot}82$ B.Th.U.,
h_b''' heat drop in the moving blades - - $1{\cdot}43$ cal., or $2{\cdot}58$ B.Th.U.,

the theoretical outlet velocity c_0''' from the third guide blades and the relative outlet velocity w_0''' from the third moving blade - - - - -	129 m. per sec., or 423 ft. per sec.,
the actual outlet velocity c_1''' from the third guide blade and the actual relative outlet velocity w_1''' from the third moving blade - - - - -	116 m. per sec., or 381 ft. per sec.,
the relative inlet velocity w''' to the third moving blade and the absolute outlet velocity c_2''' from the third moving blade - - - - - - -	69 m. per sec., or 226 ft. per sec.

From the velocity diagram the projection of the actual absolute inlet velocity to the third moving blade and of the actual absolute outlet velocity from the third moving blade is equal to 100 m. per sec., or 328 ft. per sec., and 37 m. per sec., or 121·4 ft. per sec. respectively. From the sum of these the heat drop actually consumed is $h_i''' = 2{\cdot}06$, or 3·71 B.Th.U., from which is obtained an output of 235 Ch. à Vap., or 232 H.P.

Resuming, we have for the third stage:

$c_1''' = 116$ m. per sec., or 381 ft. per sec.
$c_0''' = 129$ „ „ 423 „
$w''' = 69$ „ „ 226 „
$h_a''' = 1{\cdot}57$ cal., or 2·82 B.Th.U.
$h_b''' = 1{\cdot}43$ „ 2·58 „
$h_i''' = 2{\cdot}06$ „ 3·71 „
$\eta_i = 68{\cdot}6$ per cent.
$N_i = 235$ Ch. à Vap., or 232 H.P.

The total output of the three stages:

$$N = N' + N'' + N'' = 624 \text{ Ch. à Vap., or } 616{\cdot}5 \text{ H.P.,}$$

this being the so-called indicated output without taking account of the leakage losses.

The total thermo-dynamic efficiency ratio is:

$$\frac{h_i' + h_i'' + h_i'''}{H} = 68{\cdot}4 \text{ per cent.}$$

Calculation of the Theoretical Blade Height.

If t is the blade pitch at the mean diameter D assumed to be 10 mm.,
α the outlet angle of the blade assumed to be 30°,
δ the blade thickness (see Fig. 27) assumed to be ·5 mm., and
a the depth of blade passage,

then, $t \sin \alpha = a + \delta,$

$$a = t \sin \alpha - \delta,$$
$$a = 10 \cdot \tfrac{1}{2} - \cdot 5 = 4 \cdot 5 \text{ mm.},$$

and the number of blades:

$$z = \frac{\text{D} \cdot \pi}{t} = \frac{2{,}400 \cdot 3{\cdot}14}{10} = 754 \text{ blades.}$$

If l_a' is the blade height of the first guide blades, the required outlet area is:

$$\text{A}_a = z \cdot l_a' \cdot a,$$

and, $$l_a' = \frac{\text{A}_a}{z \cdot a} = \frac{\text{A}_a}{754 \cdot 4{\cdot}5} = \frac{\text{A}_a}{3{,}390}.$$

For the first row of guide blades $c_1' = 98$ m. per sec., or 321·5 ft. per sec. The specific volume of the steam is that of the steam in the state

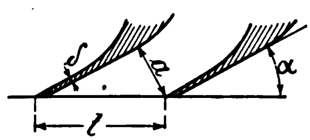

Fig. 27.

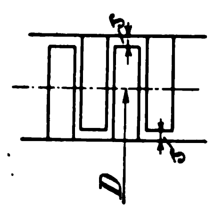

Fig. 28.

at the outlet. This can be obtained from the Mollier diagram. If the heat drop is $h_a^1 = 1{\cdot}42$ cal., or 2·56 B.Th.U., the corresponding end pressure is 1·925 atm., or 28·3 lbs. per sq. in. and the dryness factor $x = \cdot 938$, the specific volume for dry steam from the steam table will be ·937 cub. m., or 14·53 cub. ft., and the actual specific volume:

$$v_a' = \cdot 938 \cdot \cdot 937 = \cdot 879 \text{ cub. m.},$$

$$v_a' = \cdot 938 \cdot 14{\cdot}53 = 13{\cdot}62 \text{ cub. ft};$$

therefore, $$\text{A}_a = \frac{\text{Q} \cdot v_a'}{c_1'} = \frac{20 \cdot \cdot 879}{98} = \cdot 1795 \text{ sq. m.}$$

$$\text{A}_a = \quad = \frac{44{\cdot}2 \cdot 13{\cdot}62}{321{\cdot}5} = 1{\cdot}875 \text{ sq. ft.},$$

and, $$l_a' = \frac{179{,}500}{3{,}390} = 53 \text{ mm.}$$

$$l_a' = \frac{1{\cdot}875 \cdot 144}{754 \cdot \cdot 177} = 2{\cdot}02 \text{ ins.}$$

In the same manner the blade height of the guide blades of the second and third stages can be determined. On the above assumption (page 84) for the drop over each stage the blade height of the three stages will be:

$$l_a' = l_a'' = l_a''' = 53 \text{ mm.},$$

i.e., all blades of the three stages are of the same height. If the results gave different values for the three blade heights, the calculation of the blading for the three stages would have to be repeated on a different division of the heat drops.

Tip Leakage Losses and Actual Blade Height.—Both guide and moving blades have to be provided with a certain amount of clearance at the tips, and in our case we will assume that this clearance $s = 2$ mm., or ·079 in. (Fig. 28).

Due to the steam eddies in the clearance spaces the velocity of the steam over the tips will be less than the velocity through the blades, and can be assumed to be 70 per cent. of the steam velocity through the blades and equal in both guide and moving blades.

The ratio of the leakage area to the calculated outlet area from the blades is

$$= \frac{2\text{D}\pi s}{\text{A}_a} = \frac{2 \,.\, 2{,}400 \,.\, 3{\cdot}14 \,.\, 2}{179{,}500} = {\cdot}168,$$

and the leakage loss:

$$\frac{{\cdot}168 \,.\, 70}{100} = {\cdot}1176 \text{ of the quantity of the working fluid.}$$

The blade height must consequently be reduced by 53 . ·1176, and their actual height will be

$$53 - 53 \,.\, {\cdot}1176 = 47 \text{ mm.}$$

The output is reduced in the same proportion as the blade height, so that the indicated output will be:

$$\text{N}_i = 624 - 624 \,.\, {\cdot}1176 = 551 \text{ Ch. à Vap. metric units.}$$
$$\text{N}_i = 616{\cdot}5 - 616{\cdot}5 \,.\, {\cdot}1176 = 544 \text{ H.P. English units.}$$

Note.—The assumption that the steam passing through the leakage space is lost entirely is not correct, as at least part of the heat contents are utilised in the further stages.

With reference to the other losses, such as gland losses, &c., we refer to §§ 43 and 44.

§ 66. **General Remarks on the Calculation of Parsons Turbines.**—In the above we have only given the calculation for an arbitrarily selected section of the turbine consisting of three pressure stages, but as this type of turbine consists entirely of pressure stages, the example shows the method according to which the calculation would have to be carried out if no experimental data for a similar machine were available. The makers of this type of turbine naturally base their calculations mainly on data which they have obtained by experience, and the whole process of calculation is naturally much simplified.

In designing a complete installation the turbines will be so arranged that the total output is divided equally over each propeller shaft (see Part VII.). Further, the speed of the turbines will depend on the speed of the vessel, the number of shafts and the output per shaft, so that the only factors to be settled are the turbine diameter and number of expansions or stages.

On determining the diameter of the low-pressure turbine, which for reasons of efficiency will be chosen as large as possible, the available space has to be taken into consideration.

The blade height at the exhaust end is given by the mean blading diameter and the outlet area of the blade annulus; the latter depending on the pressure in the condenser and the over-all efficiency ratio of the turbine (see the heat diagram).

The final state of the steam is obtained from the Mollier diagram by dropping a vertical line from the initial pressure to the curve corresponding to the pressure in the condenser. On this line, and taking the initial pressure as the origin, a length is marked off corresponding to the heat drop, actually utilised in the turbine and depending on the efficiency ratios which have been obtained with similar designs.

From the end point of this line a horizontal line is drawn until it meets the curve of the condenser pressure. The point where this line meets the curve of the condenser pressure gives the final conditions of the steam (see § 26).

The diameter of the high-pressure turbine depends upon the consideration that the depth of the blade annulus of the first row of fixed blade, for a given initial steam condition, must not exceed a certain amount to enable the most suitable steam velocity to be obtained. Further, the blade height must not be less than a given amount, as otherwise the radial clearance becomes too large in proportion to the height of the blade.

These conditions determine the diameter as well as the blade height, both at the inlet to the high pressure turbine as well as at the exhaust end of the low-pressure turbine.

The next step is to fix the number of stages. Here again experience serves as a guide to indicate what the average heat drop per stage for a given blade velocity should be, and by dividing the total drop by the average drop per stage, the number of stages is obtained. As the blade velocities at the high-pressure end is a minimum, and on account of the proportionally large tip clearances of the high-pressure blades compared to their height, it is advisable that the heat drop over the first rows should be comparatively small. At this stage of the calculation it is advisable to draw a diagram showing the drop over each stage. On the abscissæ the stages are marked off at equal distances, the ordinate at each stage representing the heat drop for the stage. The curve should show an increasing slope towards the low-pressure end (see Fig. 29).

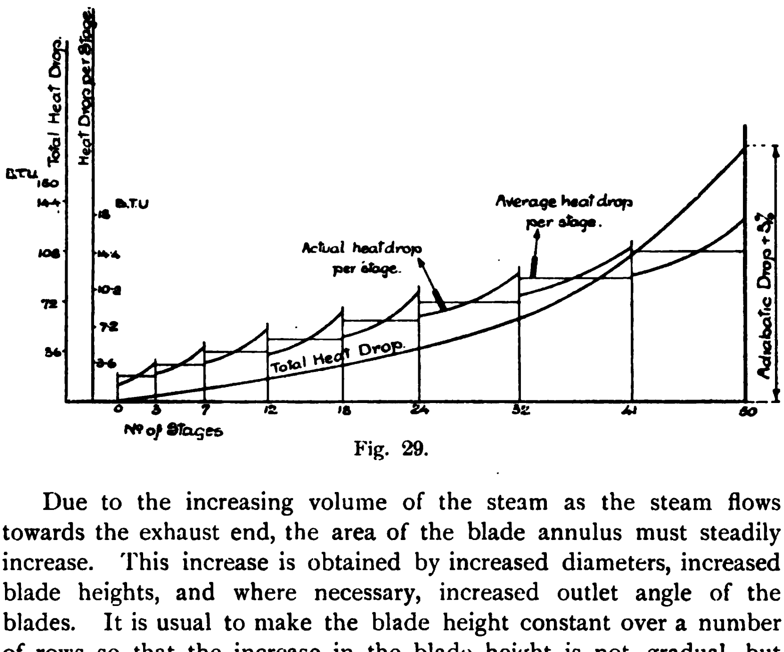

Fig. 29.

Due to the increasing volume of the steam as the steam flows towards the exhaust end, the area of the blade annulus must steadily increase. This increase is obtained by increased diameters, increased blade heights, and where necessary, increased outlet angle of the blades. It is usual to make the blade height constant over a number of rows so that the increase in the blade height is not gradual, but stepped. The result is that the drop curve in Fig. 29 will not be a continuous curve, but will be a stepped line as shown in chain line, the heat drop at the beginning of each step being below and at the end of each step above the curve which would result if each row of blades were higher than the preceding one (see Fig. 29).

PART IV.

TURBINE DESIGN.

SECTION I.

DESIGN OF THE A.E.G. TURBINE.

§ 67. **General Remarks.**—The steam turbine built by the A.E.G. has been evolved from the Curtis turbine. The high-pressure section consists of Curtis or impulse stages, each wheel being fitted with two to four velocity stages (see § 40), whilst the low-pressure section consists of a drum mainly fitted with single rows of impulse blades. The last few rows are fitted with reaction blades.

The reversing turbine is placed in the exhaust end of the turbine, and contains one or more Curtis stages and a comparatively short drum.

§ 68. **The Casing.**—In the smaller A.E.G. turbines the Curtis stages, low-pressure drum, and reversing turbine are assembled in a single casing (Figs. 30 to 32). In the larger units the turbines are divided and the Curtis stages or high-pressure turbines are entirely separated from the low-pressure drum and reversing turbine, the two latter being contained in a single casing. In some cases the last Curtis stage forms part of the drum of the low-pressure turbine. The two turbines are connected in series, the shafts being coupled by means of a flexible coupling (Fig. 33).

The steam inlet end of the high-pressure casing in which the steam is at a high pressure is usually made of cast steel, the rest of the casing being of cast iron.

The thickness of the castings at the steam inlet should be designed that the stresses in the material at the maximum pressure are about 200 kg. per sq. cm. (2,850 lbs. per sq. in.) with cast iron and about 350 kg. per sq. cm. (5,000 lbs. per sq. in.) with cast steel. The other parts of the turbine casing are dimensioned with a view to the

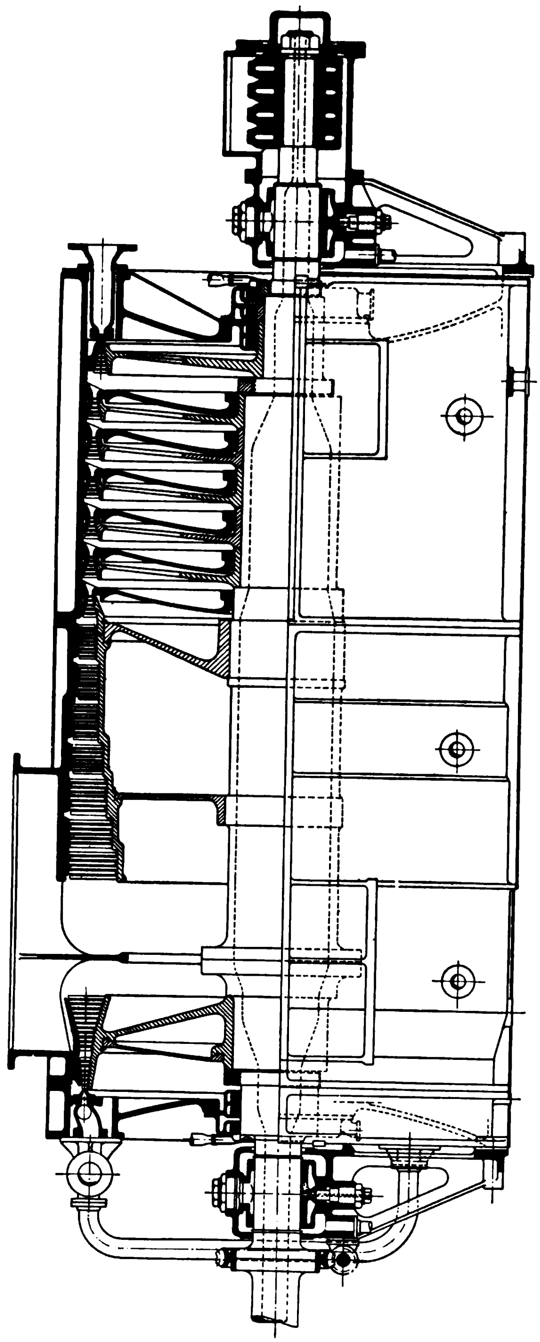

Fig. 30.

machining and handling. The casing is divided in a horizontal plane and the two halves bolted together by strong steel bolts. These are placed as close up to the body of the casing as possible so as to

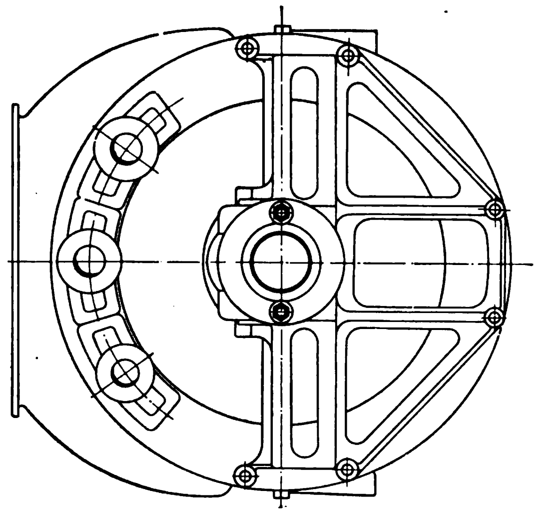

Fig. 32.

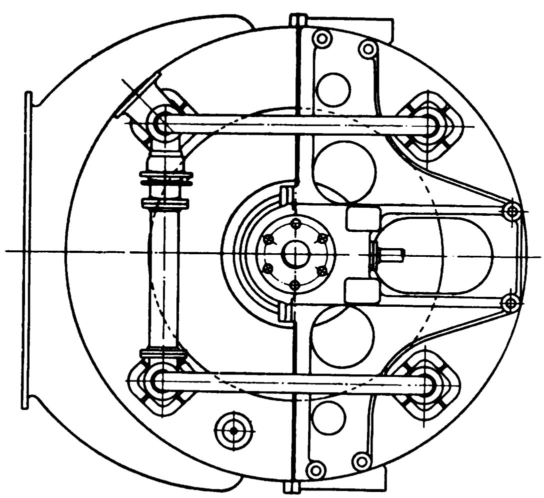

Fig. 31.

reduce the bending moment on the flanges, these being in addition strengthened by ribs. The casing is further divided into separate sections at right angles to the axis, mainly to facilitate machining and handling during erection. The flanges joining the various parts are machined and scraped, enabling a steam-tight joint to be made without any jointing material. Countersunk boltheads are used at the flange corners to enable the bolts to be placed close together. The exhaust

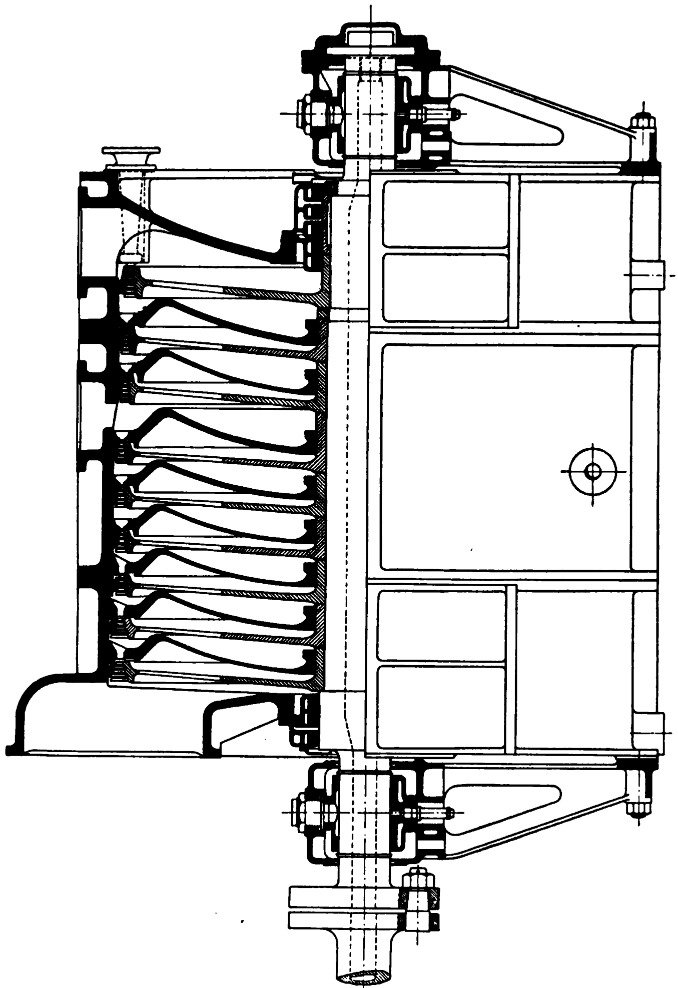

Fig. 33.

flange is placed about midway between the end of the low-pressure drum and the end of the reversing turbine, and is dimensioned for a maximum steam velocity of 100 to 150 m. per sec. (330 to 490 ft. per sec.).

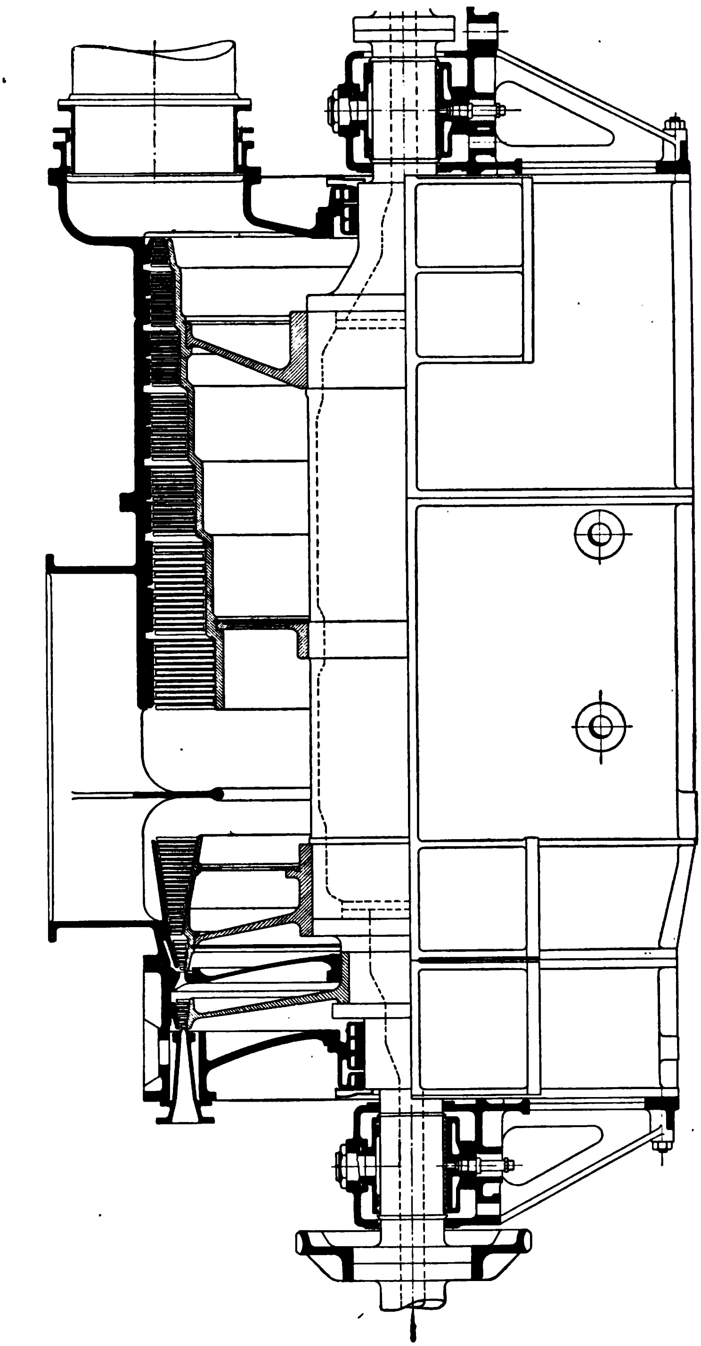

Fig. 34.

The end covers are usually cast in one with each half of the casing, and are arched inwards so as to secure maximum strength. In some cases the end covers are built up as separate units each consisting of an

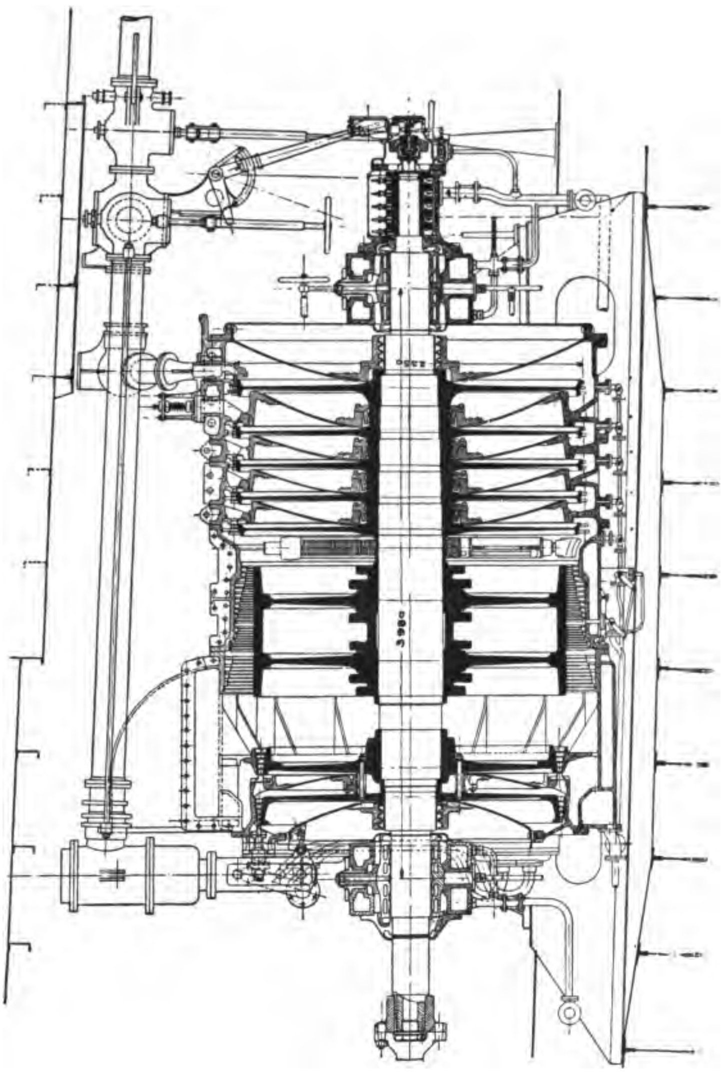

Fig. 35.

outer cast-steel ring forming the flange, and an inner ring of the same material for supporting the packing gland, both rings being riveted to a dished cover made of steel plate.

In this design a horizontal joint is not required (see Fig. 35 of S.S. "Kaiser").

The covers are fitted with pockets to take the nozzle blocks; those for the main turbines are placed in the front cover and those for the reversing turbine in the back cover.

In turbines with built-up covers as described above, the nozzle blocks are fitted to the side of the outer casing. For supporting the turbine casing and for fixing it to the foundations, feet are cast at either side at both ends of the casing. Between the supports and the foundations special packing pieces are fitted over which the feet of the supports slide in following the expansion of the casing. These supports are arranged as nearly as possible vertically below the centre of the shaft to avoid displacement of the shaft due to distortion of the casing. To allow of a free longitudinal expansion the supports are rigidly secured to the foundations at one end only, those at the opposite end being free to slide in a direction parallel to the shaft. In large units and in the low-pressure turbines of large installations, the supports nearest the exhaust end are rigidly secured. These, therefore, transmit the whole propeller thrust to the foundations and the vessel. In separate high-pressure turbines the supports astern nearest to the exhaust end are similarly secured.

In large units stops are cast on either side of the lower turbine casing, these stops fitting into corresponding foundation plates bolted to the hull of the vessel, and preventing any lateral movement of the turbine.

Grooves are turned in the cylinder casing to take the segments containing the guide blades, these segments being bolted to the casing by external studs.

§ 69. **Connections.**—In addition to the usual connections for thermometers and pressure gauges the following connections are required:—

1. A flange to take the safety valve for the first stage, set to blow off at a pressure above the maximum pressure at this stage corresponding to full output.
2. Drain connections for the steam from the glands. One leading to the second or third stage for the drainage from the inner section of the main high-pressure gland, a second connection leading to the centre of the drum for the outer section of the main high-pressure gland, and for the drainage from the other glands (see § 58).
3. Drain connections for the high-pressure stages, and for the space between the low-pressure drum and reversing turbine.

These drains must naturally be placed at the lowest point of the cylinder.

§ 70. **Diaphragms.**—Similar to the end covers of the cylinder the diaphragms between the wheels are arched to obtain a maximum stiffness. They rest at their outer periphery on the nozzle ring or segments fixed to the cylinder (Fig. 33), or on a ring cast in the cylinder, this ring also serving to support the nozzle segments. In the stages under high steam pressure the diaphragms are sometimes made with an opening to which the nozzle segments are secured, and are then supported directly from the cylinder. The diaphragms are designed to withstand the pressure between two consecutive stages

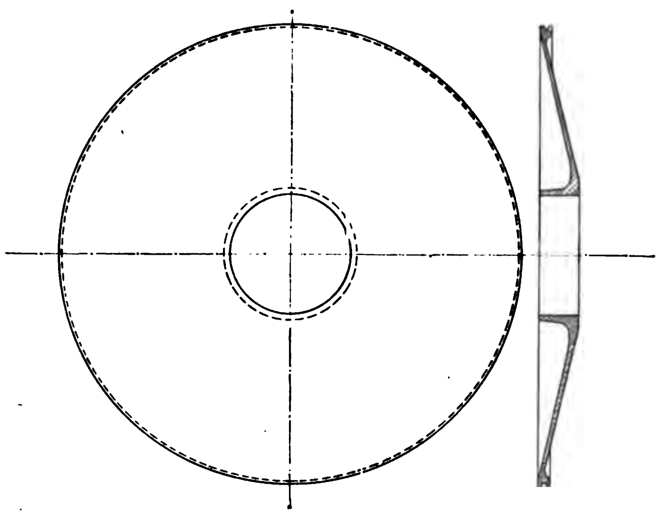

Fig. 36.

without any marked distortion. Usually they are not split, but form a single disc to economise in length, and to obtain maximum stiffness. On dismantling the machine by withdrawing the upper half of the casing they remain on the shaft. The material for the diaphragm is generally cast steel (see Fig. 36), but in some cases where it is necessary to reduce the weight of the whole machine as much as possible, the arched disc is made of a pressed steel plate riveted to a rim of cast steel or of rolled steel section (see Fig. 37).

The boss in the centre of the diaphragm is not as a rule formed to take a proper gland, but fitted with a bronze bush sometimes lined with white metal, grooves being turned in the white metal to form a labyrinth gland (Fig. 38).

The clearance between the bush and this shaft is ·2 to 1 mm. (·008 to ·04 in.) (see § 58). The smallest clearances being on the end diaphragms where the deflection of the shaft is a minimum. This

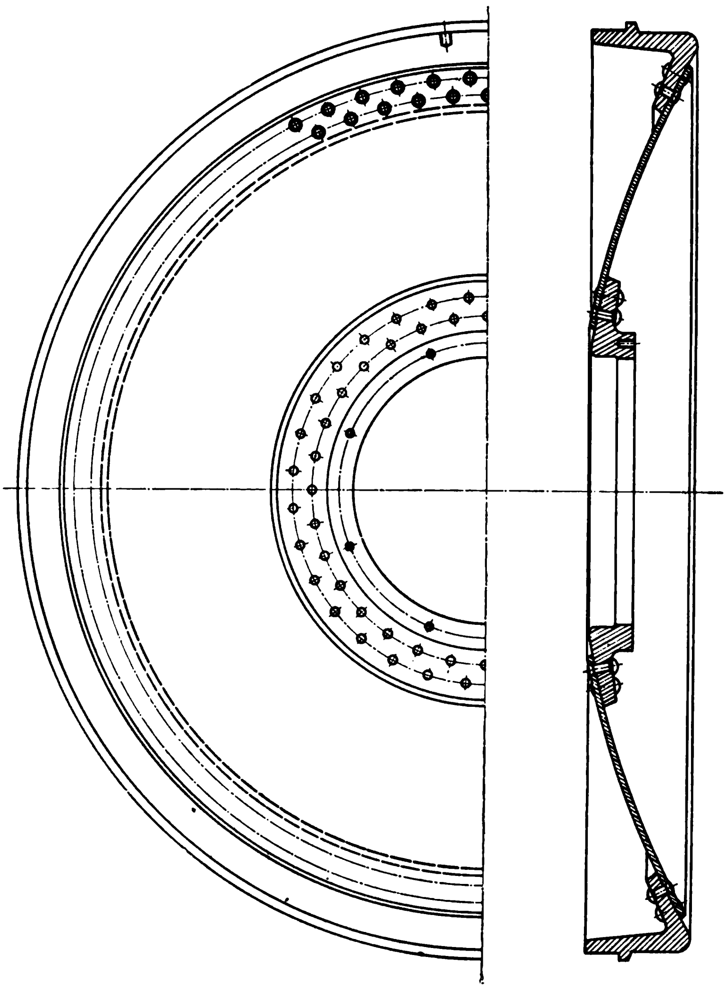

Fig. 37.

radical clearance is allowed to prevent the shaft when bent by its own weight from touching the bushes. For the same reason the position of the hole in the diaphragm is somewhat eccentric. The gland is secured to the boss of the diaphragm by four or eight locked steel tap bolts, the bush being cast with a flange or a set of lugs for the purpose.

§ 71. **Shaft.**—The shafts of marine steam turbines are generally made of ductile Siemens-Martin steel. In some cases nickel steel shafts with a low percentage of nickel are used. Figs. 39 to 41 show shafts of three different types.

Fig. 39 is a shaft which has been bored out to ascertain whether the material is sound.

Fig. 40 is a hollow shaft in two pieces, of comparatively small thickness and large diameter, to obtain a maximum stiffness with a minimum weight. The two halves are bolted together by bolts of such strength that two bolts diametrically opposite suffice to carry the shaft including the wheels with a factor of safety of 4.

Fig. 41 shows the two hollow shaft ends for a drum.

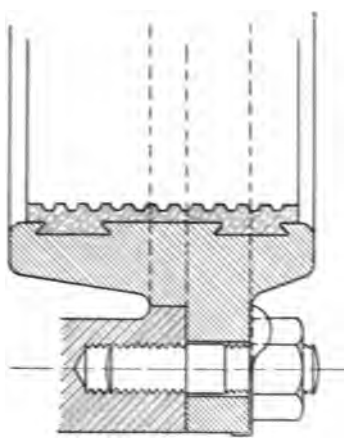

Fig. 38.

It should be made a rule that, similar with all other turbine system, the shaft is as stiff as possible, and that the calculated critical velocity (see §§ 46 to 48) is considerably higher than the maximum velocity to be expected under service conditions.

The outer diameter of the shaft is stepped to take the wheels, so that these can be easily mounted. Similarly separate flanges or sections are provided to carry the end discs of the drums. At each end of the shaft a smooth length is turned for the main glands. The front end *b* of the shaft shown in Figs. 39 and 40 projects beyond the main bearing, and serves to take the rings of the thrust block. The

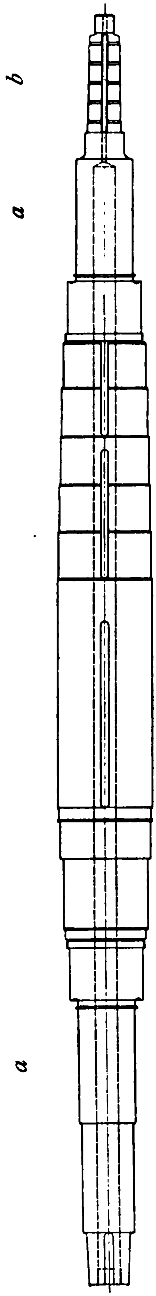

Fig. 39.

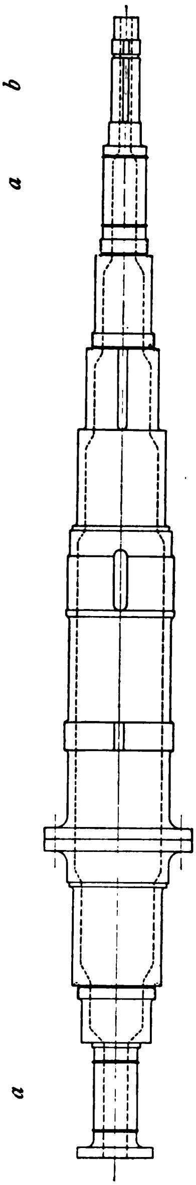

Fig. 40.

other end *a* of the shaft, like Fig. 40, is turned to take a half coupling, the end *a* of shaft, Fig. 39, being provided with a flange coupling forged in one with shaft.

The discs are keyed to the shaft to prevent rotation, and fixed longitudinally by a bronze or steel nut at the front end of the shaft (see Fig. 30).

The diameters of the shaft inside the bearings is smaller than the main shaft, and is calculated on the basis of the torsion moment of the shaft at the after end being approximately 450 to 550 kg. per sq. cm. (6,400 to 7,800 lbs. per sq. in.).

The length of the bearing is about 1·5 to 2 × diameter. The size of the bearing surface should be such that the bearing pressure is about 4 to 8 kg. per sq. cm. (57 to 114 lbs. per sq. in.). Oil throwers are turned on the shaft beyond the bearings to avoid oil being carried along the shaft and entering the glands. To the front end of the shaft the emergency governor and the tachometer drive are fitted. A

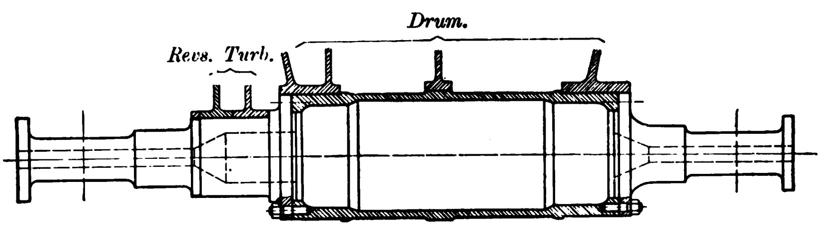

Fig. 41.

worm-wheel is fixed to the flange of the coupling for use with a barring gear.

If the dimensions of the turbine become so large that it has to be divided into a separate high and low-pressure machine, the thrust bearing is fitted to the front end of the low-pressure turbine. The two turbine shafts are connected by a short shaft rigidly connected to the low-pressure turbine shaft by a flange coupling. The other end of the connecting shaft is connected to the shaft end of the high-pressure turbine by a flange coupling with axial play, to allow of dependent expansion of the high-pressure turbine shaft. The latter shaft is provided with a single thrust ring at the front end to maintain the position of the rotor relatively to the casing.

§ 72. **Main Bearings of the Turbine.**—The main bearings similar to the bearings of the propeller shaft are only subject to small variations in pressure. The main requirement of the bearings is that the wear should be small to avoid frequent adjustment of the shaft and bearings. To obtain this result the specific pressure should be low, about 4 to 8 kg. per sq. cm. (57 to 114 lbs. per sq. in.), and the bearing bush should have a spherical seat to allow the bearing to follow the movement of the shaft. It is necessary to provide for a rapid and safe method of dissipating the heat generated in the bearing, depending on the velocity of the shaft and the bearing pressure. The velocity is as a rule considerably higher than in reciprocating engines, and it is advisable not only to provide indirect water cooling of the bearings, but forced lubrication as well. The oil forced between the shaft and the bearing surface not only maintains efficient lubrication, but as the oil is in continuous circulation it forms an excellent medium for absorbing the heat and dissipating it.

Table 11 gives the specific pressure, velocity, &c., of a number of different bearings built for marine steam turbines.

TABLE 11.

Load on Bearing.		Velocity of Journal.		Specific Pressure.	
Kg.	Lbs.	M. per sec.	Ft. per sec.	Kg. per sq. cm.	Lb. per sq. in.
3,800	8,380	5·96	19·55	6·4	91
6,750	14,900	3·85	12·64	7·8	111
10,800	23,800	4·81	15·78	3·0	42·7

Note.—All the above bearings are lined with white metal.

Figs. 42 and 43 show three views of a turbine bearing bush with spherical seating. In Fig. 42 the left half shows a longitudinal section and the right half a plan view of the lower half, and Fig. 43 a vertical section. The spheres fit into corresponding surfaces in the bearing pedestal and cover. The surfaces in the former being adjustable to enable any wear to be taken up. The two halves of the bearing bush are bolted together by four bolts. The cast-iron sleeves of the bearing bush are hollow and fitted with connections for jacketing water. The correct position of the bearing bush is registered by the oil pipe screwed into the lower half of the bush.

The bearing bushes are fixed in a pedestal secured to the two bottom ends of the cylinder (Figs. 30 to 34). The pedestals form separate hollow or ribbed castings of iron or steel bolted to the cylinder end.

The after bearing has only to support the weight of the shaft, whereas the front bearing has, in addition to carrying the weight, to

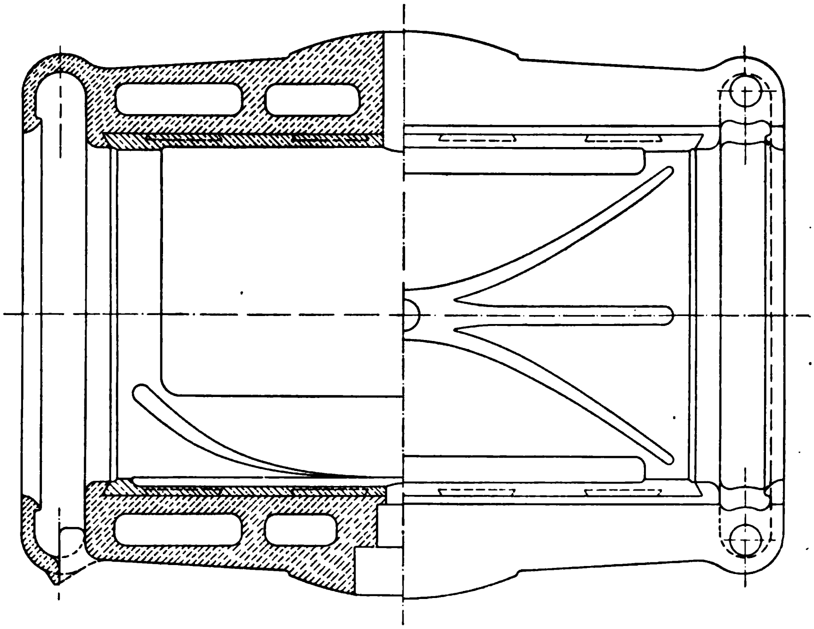

Fig. 42.

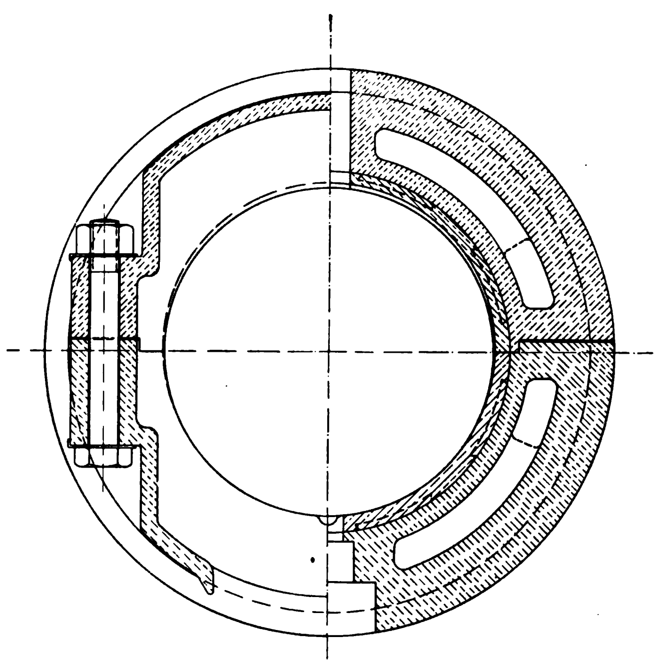

Fig. 43.

transmit the propeller thrust to the cylinder, and the bearing pedestals must be stiff enough to prevent any distortion in a horizontal plane.

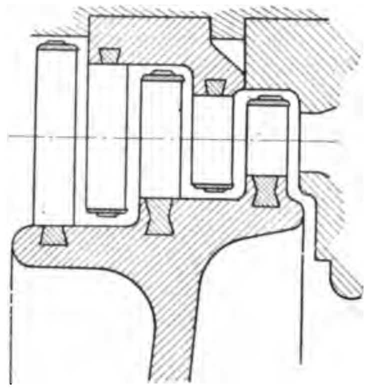

Fig. 44.

§ 73. **Construction of the Wheel and Drum.**—The material of the wheels is cast steel, except for very high velocities, when forged steel is used. The wheels are usually flat or slightly coned discs. With coned discs the diaphragms can also be slightly conical, thereby

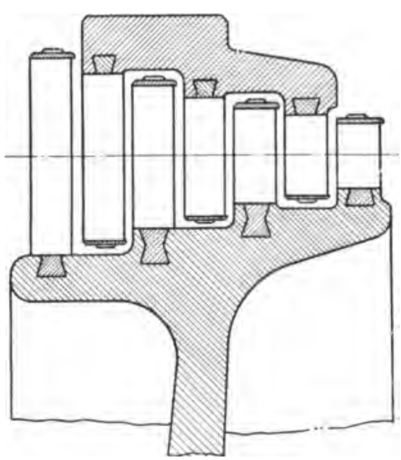

Fig. 45.

increasing their strength and maintaining the same length of each stage.

The thickness of the disc at the boss is about twice to three times the thickness at the rim. Six or eight holes are drilled in the disc to equalise the steam pressure on either side. The edge of these holes should be rounded or sharpened to diminish ventilating losses.

The rim of the discs is about 20 to 25 mm. (·79 to ·985 in.) thick, and provided with grooves of a depth of about 8 to 10 mm. (·315 to ·394 in.), to take the blade roots (Figs. 44 to 46). The rim can be arranged symmetrically relatively to the disc (Figs. 44 and 45), or to one side as shown in Fig. 46. The diameter of the boss is kept as small as possible to minimise the leakage losses through the intermediate glands, and for this reason the thickness of the boss is about 25 to 50 mm. (·985 to 1·97 in.).

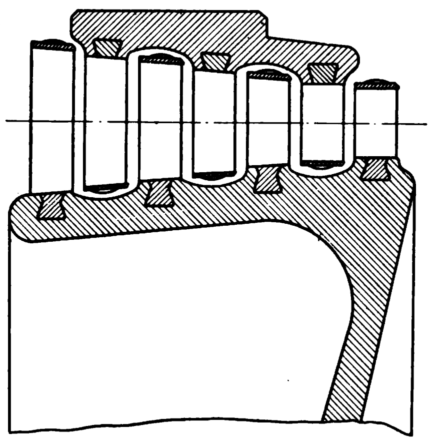

Fig. 46.

The discs are keyed to the shaft (see § 71). Utmost care in manufacture is necessary to ensure a good fit of the discs on the shaft, and of the bosses one against the other, so that all the discs are at right angles to the shaft. The boss of each wheel is provided with two tapped holes to facilitate the withdrawal of the wheels for inspection of the intermediate glands.

The drum of the main turbines and of the reversing turbines of large units consists of a cast or forged steel cylinder supported at each end by a disc of the same material. The front disc is similar in design to the turbine discs, but of a stiffer construction to prevent any longitudinal movement of the drum. The disc at the exhaust end has only to support the weight of the drum, and is made sufficiently elastic to allow of longitudinal expansion of the drum. It is therefore of a much lighter construction than the front disc.

Figs. 30 and 34 show the method of securing the drum to the discs. After the drum has been mounted on the discs, ring-shaped steel keys are caulked into the grooves turned into the rim of the discs, and the drum in addition pinned to the discs.

The discs are forced on to and keyed to the shaft. In small units the drum of the astern turbine is made in one with the conical disc (Fig. 30), and stayed by means of a second conical disc secured to the boss and the drum.

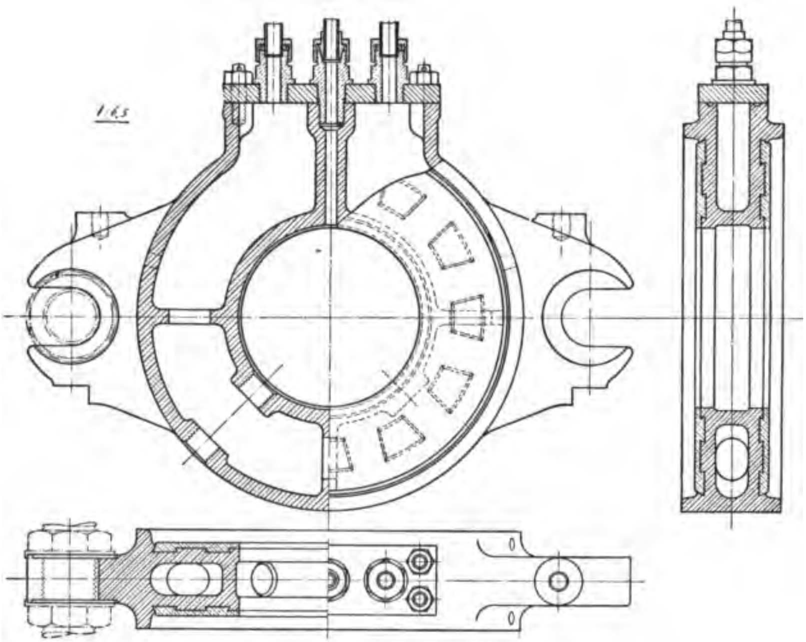

Fig. 47.

§ 74. **Thrust Bearings.**—In small turbines consisting of a single unit the thrust bearing is fitted to the forward end of the cylinder casting. In large sets with separate high and low-pressure turbines the thrust bearing is fitted to the forward end of the low-pressure machine. In the former the shaft is lengthened at the forward end, and a number of thrust rings with elongated bosses serving as distance pieces are threaded on to the shaft. The rings are prevented from turning by a key, whilst they are prevented from moving in an axial direction by a nut on the end of the shaft. The stationary thrust collars are hollow steel or bronze castings, the bearing surfaces being faced with white metal in the ordinary way. At diametrically opposite sides two lugs are cast on to the thrust collars to slide over the side rods (see Fig. 47). In the thrust bearing shown in Fig. 48 the thrust rings are maintained in position by distance pieces and locked by two keys in the outer casting.

If the thrust bearing is placed on the forward low-pressure bearing block the thrust shaft is formed by a separate length of shafting, and

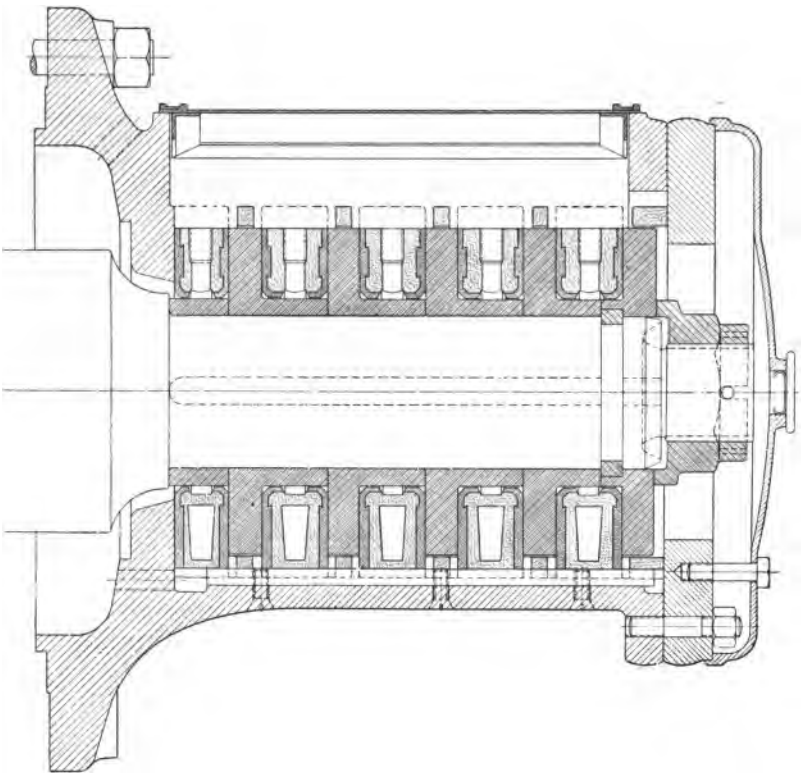

Fig. 48.

the thrust rings are in the shape of a horse-shoe similar to the arrangement used in reciprocating marine engines. The lugs supporting the horse-shoe caps on the side rods are so arranged that the line connecting the centres of the rods passes through the centre of gravity of the thrust surface. The thrust collars are secured to the rods by means of nuts, these transmitting the propeller thrust or the thrust on the drum due to the steam, to the two rods and to the main thrust block casting which is bolted to the main bearing block. The thrust collars are lubricated with oil under a pressure of 1 to 4 atm. (approx. 15 to 60 lbs.). Cooling water from the circulating water mains is supplied in the usual manner to the hollow casting of the thrust collars.

It may be of interest to give the results of tests made by the A.E.G. on the thrust bearings of the S.S. "Kaiser." *

The bearing was designed for a maximum thrust of 50 tons (110,000 lbs.) at a maximum number of revolutions of 900 per minute.

The thrust rings were of steel and five in number. The thrust collars were hollow gun-metal castings arranged for water cooling.

A preliminary test showed the most suitable type of thrust bearing design to be that shown in Figs. 48 to 50. The thrust collars had a

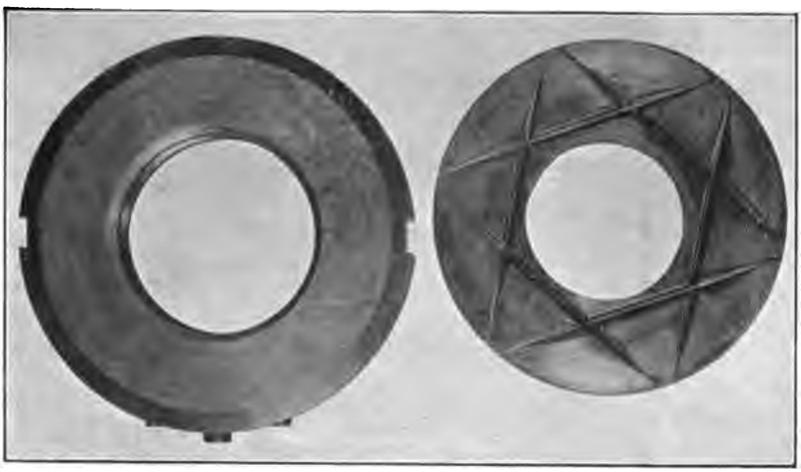

Fig. 49. Fig. 50.

smooth white metal surface without oil grooves. On the hardened thrust rings oil grooves were milled approximately tangential to the inner circumference of the rings.

The oil was introduced through the thrust collars at the inside circumference at a pressure of about 1 to 2 atm. (15 to 30 lbs. per sq. in.).

The bearing was capable of being highly loaded, a single thrust collar carrying a load of 15,000 kg. (33,000 lbs.) at 900 rev. per min. Figs. 49 and 50 show the rings after the thrust was increased to 12 tons = 12,000 kg. (26,500 lbs.). An examination of the rings showed

* See "Zeitschrift des Verrius Deutsche Ingenieure," 1906, p. 1358.

practically no signs of wear of the white metal bearing surface, the tool marks being still visible and only very few circular marks made by small particles of dust introduced by the oil. Pure "Valvoline" as well as a mixture of two parts "Valvoline" to one part of rape-seed oil were used as a lubricant.

The latter mixture appeared to have a greater tendency to foam than the pure oil.

A number of tests were made, the power absorbed by the rings at different speeds being measured by the temperature rise of the oil and cooling water.

The maximum loading on the surfaces was as follows:—

Outside diameter of bearing surface	298 mm.	11·7 in.
Inside " " "	185 mm.	7·3 in.
Area of oil grooves - - -	47 sq. cm.	7·285 sq. in.
Nett bearing surface - - -	382 sq. cm.	59·2 sq. in.
Diameter of the circle of centre of gravity of bearing surface -	246 mm.	9·7 in.
Revolutions per minute - -	900	900
Total thrust - - - -	15,000 kg.	33,000 lbs.
Specific thrust - - - -	39·1 kg. per cm.	555 lbs. per sq. in.
Outside circumferential velocity -	14 m. per sec.	45·9 ft. per sec.
Velocity of centre of gravity -	11·6 m. per sec.	38·06 ft. per sec.
Power absorbed at no load -	4·2 H.P.	4·2 H.P.
" " at maximum load	12 H.P.	12 H.P.

Below are some figures giving details of some thrust bearings in operation.

TABLE 12.

Total Thrust.		Velocity.		Specific Thrust.	
Kg.	Lbs.	M. per sec.	Ft. per sec.	Kg. per sq. in.	Lbs. per sq. in.
11,000	24,250	7·1	23·29	4·66	66·27
7,300	16,094	5·66	18·56	8·65	122·8
15,000	33,069	5·82	19·09	4·15	58·9

All the above had white metal bearing surfaces.

§ 75. **Main Steam Glands.**—Where the shaft passes through the casing stuffing boxes or glands are provided. The design of these glands is shown in Fig. 51. The gland case is usually of brass and split in a horizontal plane to facilitate dismantling and inspection. For the same reason the casing is sometimes divided in two halves longitudinally to allow the distance between the turbine and the bearing

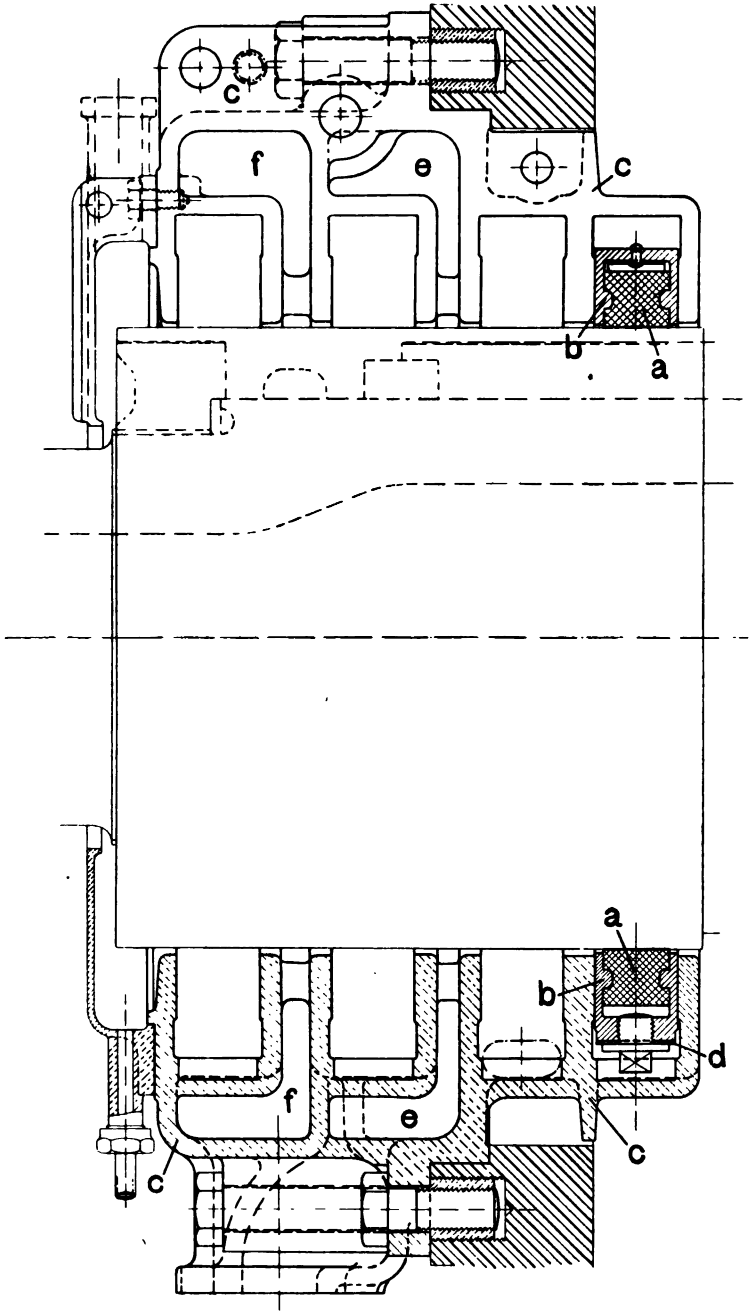

Fig. 51.

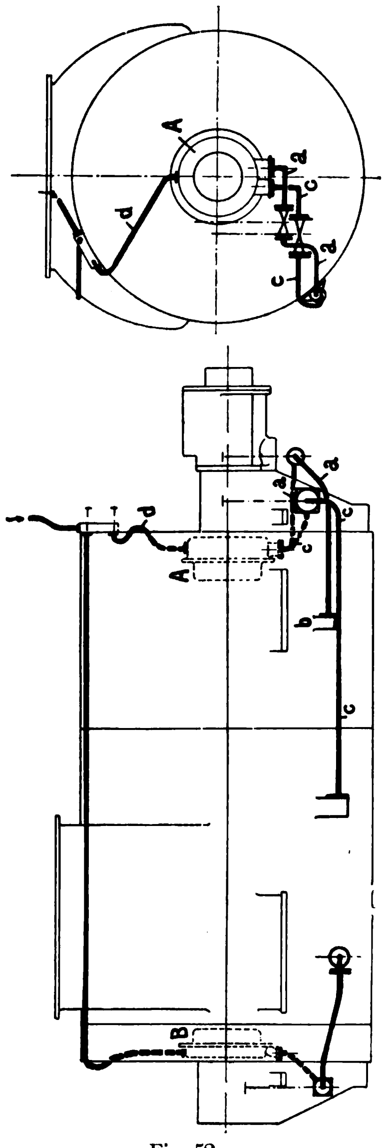

Fig. 52.

block to be made as short as possible, still giving sufficient room for dismantling the gland.

Forward Gland.—This gland is denoted by A in Fig. 52, and a sectional view is given in Fig. 51. A brass casting C is fitted to the cylinder end concentric to the shaft. Cast-iron channel shaped rings *b* fit into the spaces in the casing and serve as carriers for the carbon packing rings *a*. The cast-iron rings *b* are in three sections, and the weight of these is taken by the annular spring *d*. The packing ring is thus enabled to follow freely all changes in the shaft diameter due to the influence of temperature. Pockets *e* and *f* are formed between the rings. The inside pocket *e* is connected by the "leak-off" pipe *a* (Fig. 52) to a lower expansion of the turbine (at *b* in Fig. 52). This "leak-off" connection is provided with a cock, and is only used if the steam leakage through the gland should become excessive through wear of the packing rings.

Under ordinary running conditions the second "leak-off" connection *c* (Fig. 52) only is used. This pipe *c* connects the gland pocket *f* (Fig. 51) to the low-pressure turbine. An automatic valve is fitted in this pipe set to maintain a pressure in the pocket *f* slightly above the atmosphere.

A steam connection *d* (Fig. 52) is provided to the pocket *f* (Fig. 51) to admit boiler steam to the gland. This is necessary to prevent air leakage should the high-pressure turbine be running in vacuum when steaming at reduced speed or when steaming astern. The valve in this pipe connection must be adjusted so that there is always a small amount of steam escaping to the atmosphere from the gland to show that there is sufficient steam pressure on the gland to prevent air entering the glands.

Glands at After End.—The gland B in Fig. 52 is always under condenser pressure when the turbine is running ahead or under a low steam pressure when running astern, and is fitted with a smaller number of carbon packing rings than the forward gland A. The construction is the same as that of the forward gland, but only one "leak-off" is provided connected to the turbine exhaust.

§ 76. **Blading of the A.E.G. Turbine.**—The wheels of the Curtis stages are fitted with two or four velocity stages. The steam admission to the first wheel from the nozzle segment is partial. The segment over which steam is admitted becomes greater in the stages towards the exhaust end proportionally to the increase in the volume of the steam, until at the last stage the steam admission occurs over the whole circumference. The last Curtis wheel is usually fitted to the front end of the drum, thus saving a disc (Figs. 30 and 34).

Fig. 53 shows the *Nozzle Box* and the group of nozzles through which the boiler steam is admitted to the first wheel. The nozzle box is of cast steel, iron, or brass, bolted to the side of the cylinder or to the cylinder end cover. The guide blades are of nickel steel and fixed in a brass casting bolted to the nozzle box. If the pressure in the first stage is below the critical pressure the nozzle has to be of the expanding type (see § 27), and the nozzle blades are then made of brass cast in one with the brass nozzle plate (Fig. 54) in place of a box.

The *Guide Blades and Moving Blades* of the Curtis stages are of

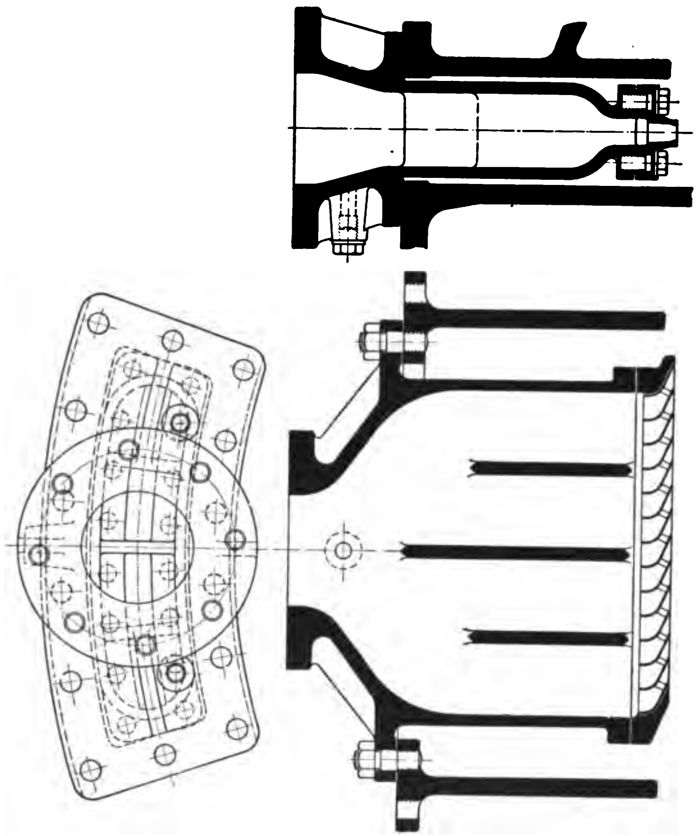

Fig. 53.

drawn brass, or if superheated steam is used of nickel steel. The roots of the blades are milled to a conical shape and fixed in dovetailed grooves. The width of the blade is 15 to 20 mm. (·59 to ·787 in.), and the axial blade clearance about 4 to 10 mm. (·157 to ·393 in.). The outer and inner ends of the blades are finished cylindrical as shown in Figs. 44 and 45, or, more correctly, in a conical manner (Figs. 46 and 55).

The distance pieces between the roots of the blades are of soft brass to enable them to be caulked after they are fitted in position. To protect the tips of the blades and limit the height of the blade passage a shrouding strip is riveted to the blade tips. The guide blades between two adjacent rows of moving blades are made in segments of

Fig. 54.

cast steel, or are of forged steel fixed in a similar manner to the moving blades. The individual blade segments are only made as long as the segment of the wheel over which the steam is admitted by the nozzles. These segments are bolted to the cylinder casing from outside. The nozzles in the diaphragms are either of the same type as the high-pressure nozzles, and fitted to openings in the diaphragms, or are

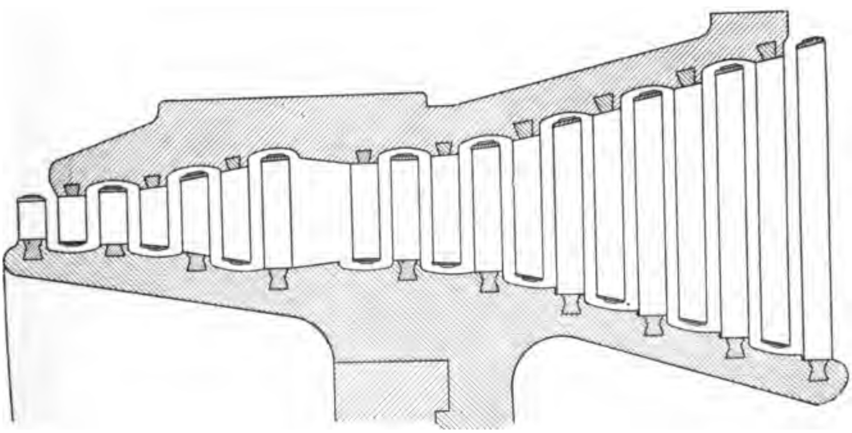

Fig. 55.

formed by separate rings of cast iron with nickel steel blades bolted to the casing (Fig. 33).

The *Blades in the Drums* are fitted in the same manner as those in the discs. They are shrouded with a brass band at the blade tips. The fixed blades are fitted in grooves turned in the cylinder casting, or in separate semicircular segments of cast steel containing four or six rows of stationary blades, and bolted in grooves to the cylinder casing.

The *Reversing Turbines* are designed in the same manner as the main turbines, with the sole difference that they contain a smaller number of rows of blades.

The steam admission of the nozzle box to the first disc or row of blades is much more extended than in the main turbine, and practically extends over the whole circumference. The guide blades of the drum section are usually fixed to a separate cast-steel sleeve bolted to the cylinder casting (see Figs. 35).

SECTION II.

DESIGN OF THE CURTIS TURBINES.

§ 77. **General Remarks.**—The Curtis turbine is a multi-stage impulse turbine, the wheels of which are fitted with velocity stages, each wheel rotating in a separate chamber. The design of the turbine is practically the same as the high-pressure section of the A.E.G. turbine, and is fitted both in the main and reversing turbine with wheels with two to four velocity stages, both turbines being arranged in a single casing.

§ 78. **Turbine Casing.**—This is made of cast iron, as the pressure in the first stage is moderately low. The casing is divided in a horizontal plane, the two halves being bolted together in the usual way. Large casings are divided in a vertical plane to facilitate machining. The arched casing ends are of cast iron, and bolted by flanges to the cylinder. The nozzles for the main and reversing turbines are bolted to the casing ends (see Fig. 56). The common exhaust flange for both turbines is placed at the top, as the condenser is generally mounted at a higher level than the turbine. The whole turbine rests on the foundation in a similar manner as the A.E.G. turbine.

§ 79. **Diaphragms.**—These are arched and are in one piece, the material used being either cast steel or cast iron. The outer rim is either flanged and bolted to the casing, or the outer rim fits into grooves in internal flanges of the casing. These internal flanges also carry the nozzle plates or boxes (see Fig. 56).

§ 80. **Glands, Shaft, and Bearings.**—These parts are identical with those described above for the A.E.G. turbine. The thrust block is fitted at the forward end.

§ 81. **Wheels.**—In large units the wheels are built up of steel discs riveted to cast-steel bosses and rims.

§ 82. **Blading.**—The nozzles are formed of nickel steel blades cast into the nozzle plates and bolted to the end covers or diaphragms.

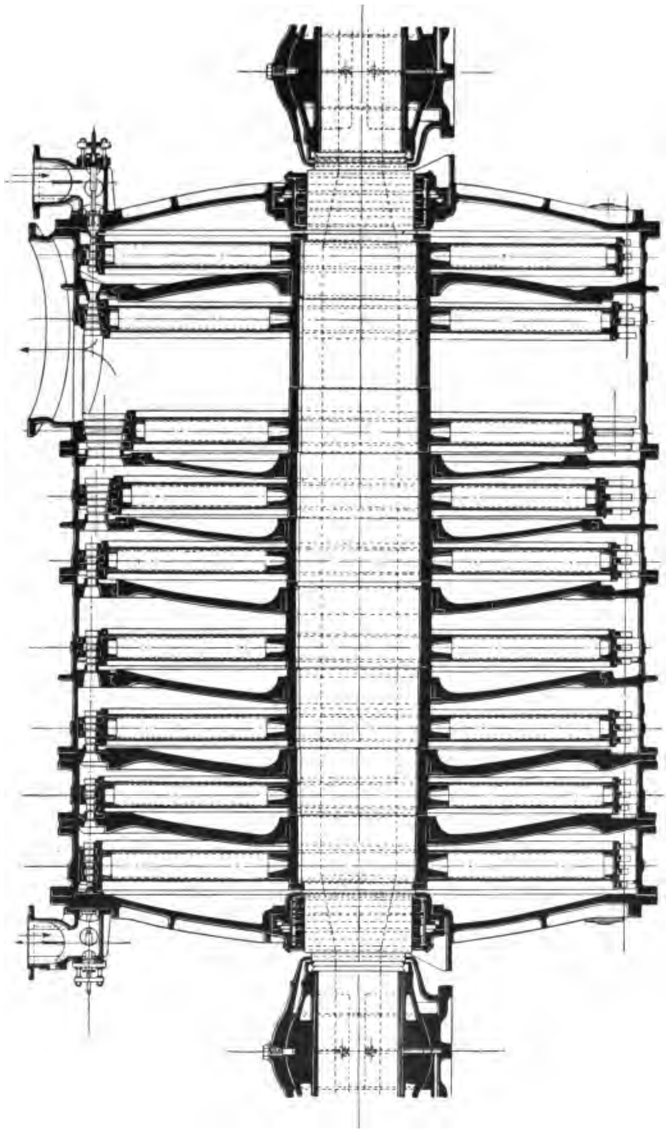

Fig. 56.

The rotating blades are sometimes cast with their roots into segments or riveted together at the roots to form segments, and fitted into the wheel rim and caulked home. The axial clearance amounts to 3 to 6 mm. (·118 to ·236 in.) to allow for unequal expansion of the casing and rotor. The guide blades are constructed on the same lines.

§ 83. **Steam Admission.**—The high-pressure nozzles are divided into groups, the steam admission to each group being controlled by a valve in the steam admission chamber.

SECTION III.

DESIGN OF THE PARSONS TURBINE.

§ 84. **General Remarks.**—The Parsons turbine is a reaction turbine in which the pressure is converted into velocity both in the fixed and rotating blades (see § 65). The turbine consists of bladed drums with full steam admission from the steam inlet to the outlet.

Each row of blades converts a small heat drop into mechanical energy, so that a large number of rows are required to utilise the whole heat drop between the boiler and exhaust pressure. To obtain the best possible effect a long turbine is necessary, and the inventor therefore subdivides the turbine into a number of units, each unit driving separate shafts, the steam passing through each unit in succession.

When steaming at slow speeds running separate turbine units in series does not allow a sufficiently high efficiency to be obtained, and for cruising speeds one or more cruising turbines are connected in series with the high-pressure turbine. Fig. 86, p. 179, shows the arrangement of the turbines in a small cruiser with four propeller shafts.

At full speed the steam first passes through the two high-pressure turbines driving the two outer shafts, and from the exhaust of the high-pressure turbines the steam is led to the low-pressure turbine on the centre shaft. At cruising speeds the steam passes through the high-pressure turbine on the centre starboard shaft, then through the low-pressure cruising turbine on the centre port shaft, and from here through the high-pressure and low-pressure main turbines to the condensers.

As shown in Part VII., reversing turbines are arranged by Parsons either on all shafts or on two of the shafts only. In the latest quadruple shaft arrangement the high-pressure reversing turbines are arranged on the two outer shafts, and the low-pressure reversing turbines on the two inner shafts.

In most cases the low-pressure reversing turbine is combined with one of the main low-pressure turbines, which has the advantage of only one exhaust pipe for both turbines being necessary.

The following illustrations show the Parsons turbines of the marine type:—

Fig. 57. A longitudinal section of a high-pressure turbine.

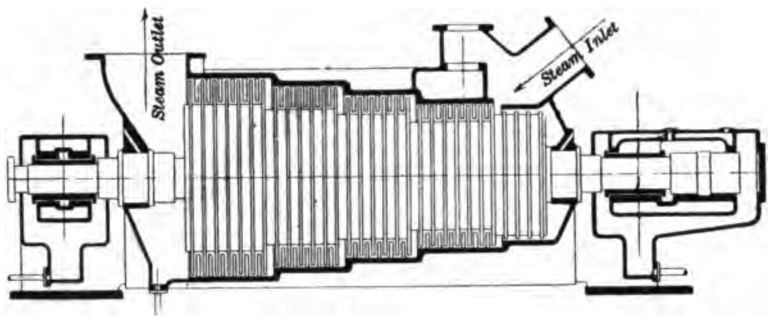

Fig. 57.

Fig. 58. A longitudinal section of the low-pressure and reversing turbine.

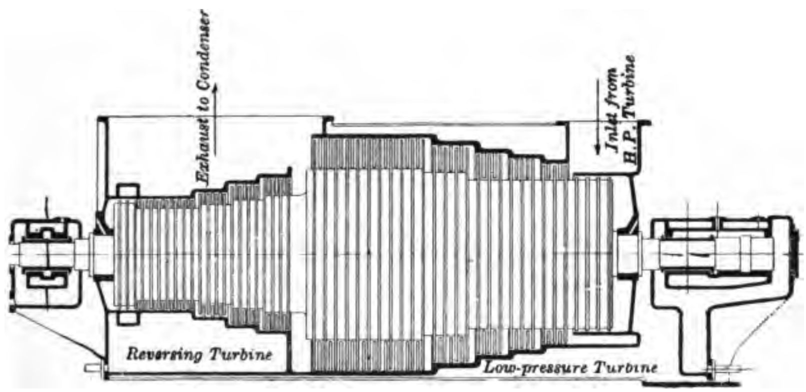

Fig. 58.

Fig. 59. A longitudinal section of the cruising turbine.

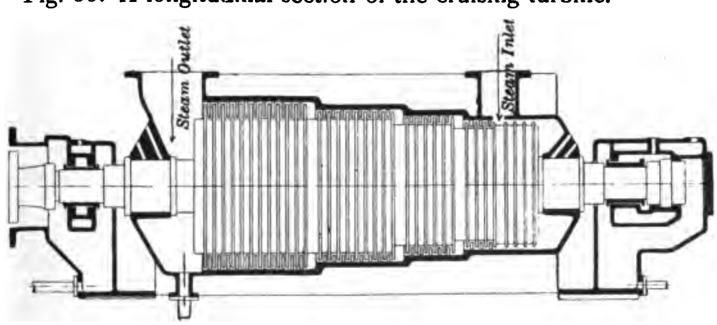

Fig. 59.

§ 85. **The Turbine Rotor.**—The hollow drums of the rotor are usually made of cast steel, or forged or rolled Siemens-Martin steel, and are connected to the spindle ends by cast-steel spiders.

The shafts do not pass through the drum, but form short spindle ends which are shrunk into the spiders. The design naturally increases

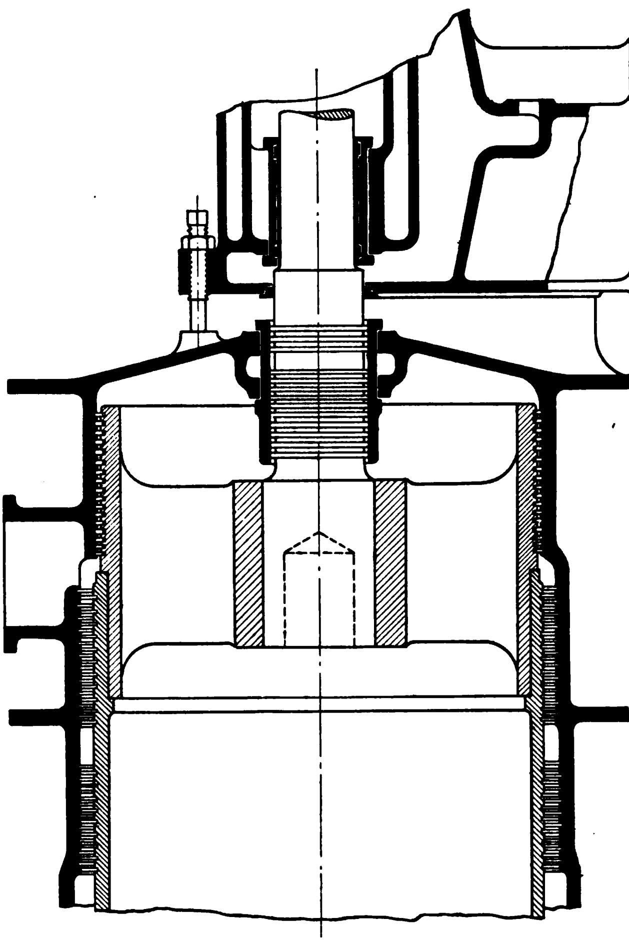

Fig. 60.

the stress on the spider, but at the same time saves considerable weight as compared with a continuous shaft through the drum.

The drum must be closed at the inlet end to the passage of the steam, and this is done by forming the end of the drum as a dummy piston. These are made by turning a number of narrow grooves in the

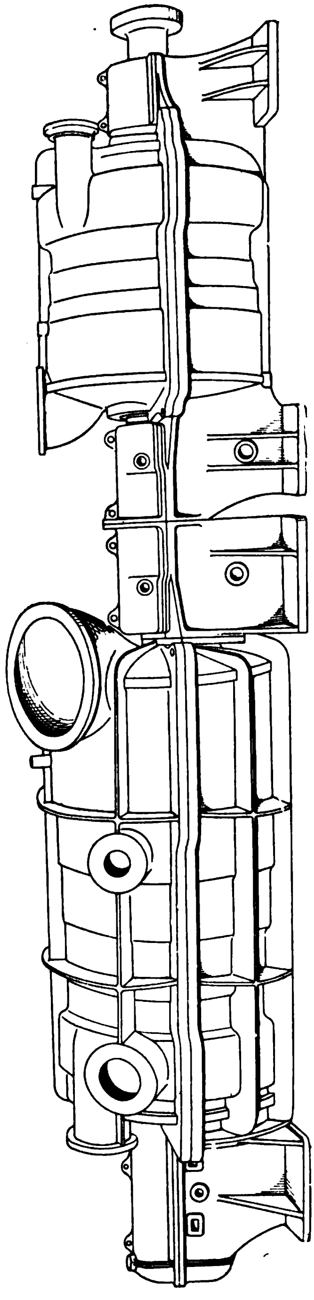

Fig. 61.

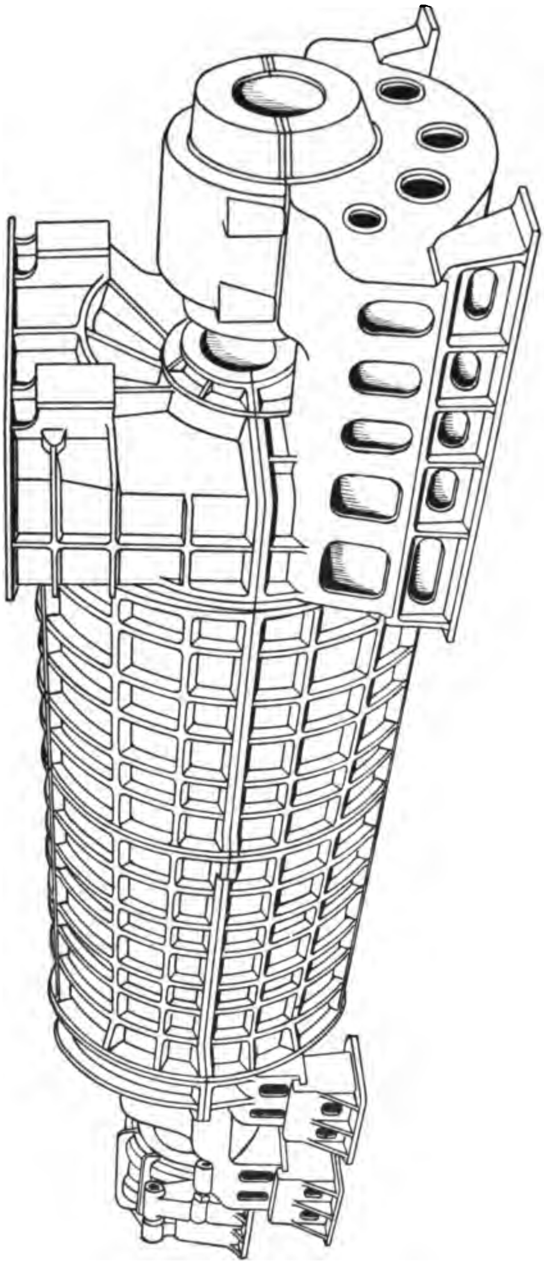

Fig. 62.

drum and cylinder into which rings are caulked, forming a labyrinth packing. The over-all diameter of the dummy rings is about the same as that of the bladed drum, and is calculated so that the steam pressure on the dummy rings compensates the propeller thrust and reaction pressure of the turbine. If the diameter of the dummy rings is smaller than the diameter of the rotor, then a thrust will be exerted equal to the annulus formed by the outer blade ring and the dummy rings multiplied by the steam pressure (see Thrust Bearings below). The spindle ends are of forged Siemens-Martin steel, and to save weight in marine turbines are hollowed when possible.

§ 86. **Bearings.**—The bearing bushes are of bronze lined with special white metal, and supported in cast-iron or steel casings. The bearings themselves are arranged outside the turbine casing so as to be easily accessible. In smaller turbines the bearing supports are cast in one with the bottom half of the turbine casing (see Fig. 61). In very large units these supports are separated from the turbine casing (see Fig. 62).

Forced lubrication is used, the oil being supplied by separate pumps, and after passing through the bearings returned in the usual manner (see § 96).

In the latest designs of Parsons, water jacketing of the bearings is not used, as the forced lubrication of the bearings alone has proved sufficient.

§ 87. **Thrust Bearings.**—Due to the reaction in the Parsons turbine, a pressure is exerted by the steam in the direction of the flow of steam, tending to displace the turbine rotor in that direction.

If the steam flows aft, this pressure acts in opposition to the propeller thrust, and the thrust bearing has only to take up the difference between the reaction pressure and the propeller thrust. By a suitable choice of the diameters of the dummy piston, it lies in the hands of the designer to influence the unbalanced thrust. It is consequently necessary to ascertain the thrust due to the reaction and the propeller thrust at various speeds and loads, and to design the thrust bearing according to the maximum resulting thrust.

The thrust bearing of the Parsons turbine consists of a number of brass rings which are fixed alternately in grooves in the shaft and grooves cut in the bearing bush. The two halves of the bearing bush are displaced so that the upper half of the rings takes up the ahead thrust, and the lower rings the astern thrust. The displacement of the two bearing halves is done by screws and keys, enabling the bearing to be adjusted to obtain the correct amount of contact between the moving and fixed rings of the thrust bearing (see Fig. 63). Oil from the general lubricating system of the turbine is used for lubrication. On account of the small longitudinal clearances between the fixed and moving blades, the correct adjustment of the thrust bearing is of the utmost importance, and the expansion of the rotor and turbine casing has to be allowed for in making this adjustment.

§ 88. **Glands.**—The shaft passes through glands in the ends of the turbine casing. These are of the Parsons labyrinth type, and are under a pressure equal to the pressure of the steam inside the turbine casing end. Two types of glands have to be considered.

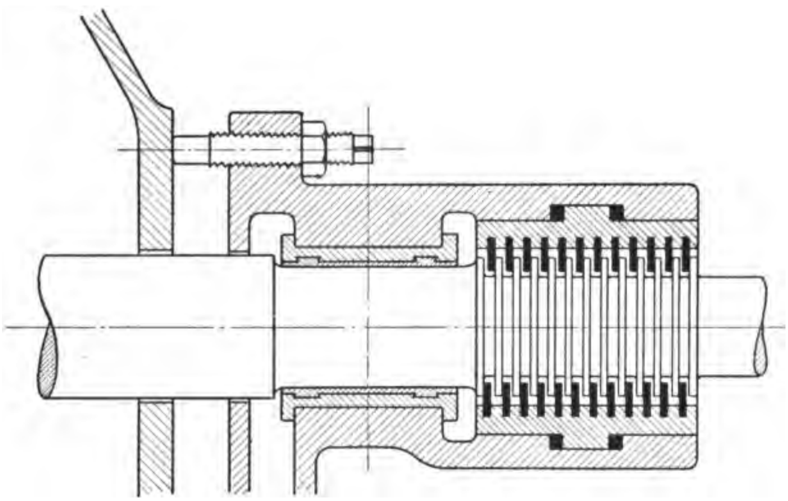

Fig. 63.

A. Such glands which are under a pressure of about 1·2 atm. (about 18 lbs. per sq. in. absolute), or under a pressure corresponding to the vacuum in the cylinder. These glands are fitted with loose rings resting in grooves turned in the spindle. This type is used in the high-pressure and low-pressure main turbines.

B. Glands which have to act against a higher inner pressure of about 5 atm. (75 lbs. per sq. in. absolute), or against atmospheric pressure. These consist of separate sleeves fixed in the casing end and fitted with a number of rings surrounding the shaft, the inner periphery of the rings projecting into grooves turned in the shaft. These glands are used in cruising turbines.

High-pressure steam is taken to the glands, type A. The amount of steam is adjustable, so that a small amount of steam is allowed to escape from the gland.

The glands, type B, are fitted with an outer and also with an inner connection. With vacuum in the cylinder, high-pressure steam is admitted to the glands, as under A, and the valve connecting the inner gland is closed. At a higher internal pressure, that is, with the cruising turbines in action, the steam passing through the gland is led through the inner connection by means of a separate pipe to the exhaust space of the high-pressure turbine. The outer connections are then connected direct to the condenser, so that any leakage steam not passing from the inner port to the exhaust of the high-pressure main turbine is led to the condenser. The valves are again adjusted, so that a small amount of steam escapes to the atmosphere to indicate that the gland is in operation. The glands of the high-pressure main turbines are connected by a separate pipe line to the condenser, so that any steam passing through the gland is led away to the condenser.

§ 89. **Turbine Casing.**—The shape of the casing or cylinder of the Parsons turbine is very simple. The casing is divided in a horizontal plane into an upper and lower half, which are securely bolted together. Although the shape is a simple one, the design has to be most carefully thought out, and the following points kept in mind:—

1. Due to the circumferential stresses under steam at high pressure and superheat, the flanges of the joint must not only be made heavy, but the transition from the flange to the cylinder must be gradual.

2. The design must ensure a uniform expansion to avoid distortion in the longitudinal direction and to prevent the casing warping into an oval shape. This is a most important point in view of the small blade clearances.

To prevent unequal expansion the casing is usually ribbed, but special care in the disposition and dimensioning of the ribs is necessary, and high ribs especially must be considered a designer's error. (Figs. 62 and 64.)

Fig. 64 is an illustration of the cast-iron turbine casing of the low-pressure turbine of the "Mauretania." In naval turbines the casing of turbines subject to high-pressure steam, such as the high-pressure and cruising turbine, is usually made of cast steel. The thickness of those parts which do not carry any blades is made equal to the thickness of the cylinders of reciprocating engines for the same type of vessel. The thickness of the parts containing the blading is increased by an amount dependent upon the size of the blades.

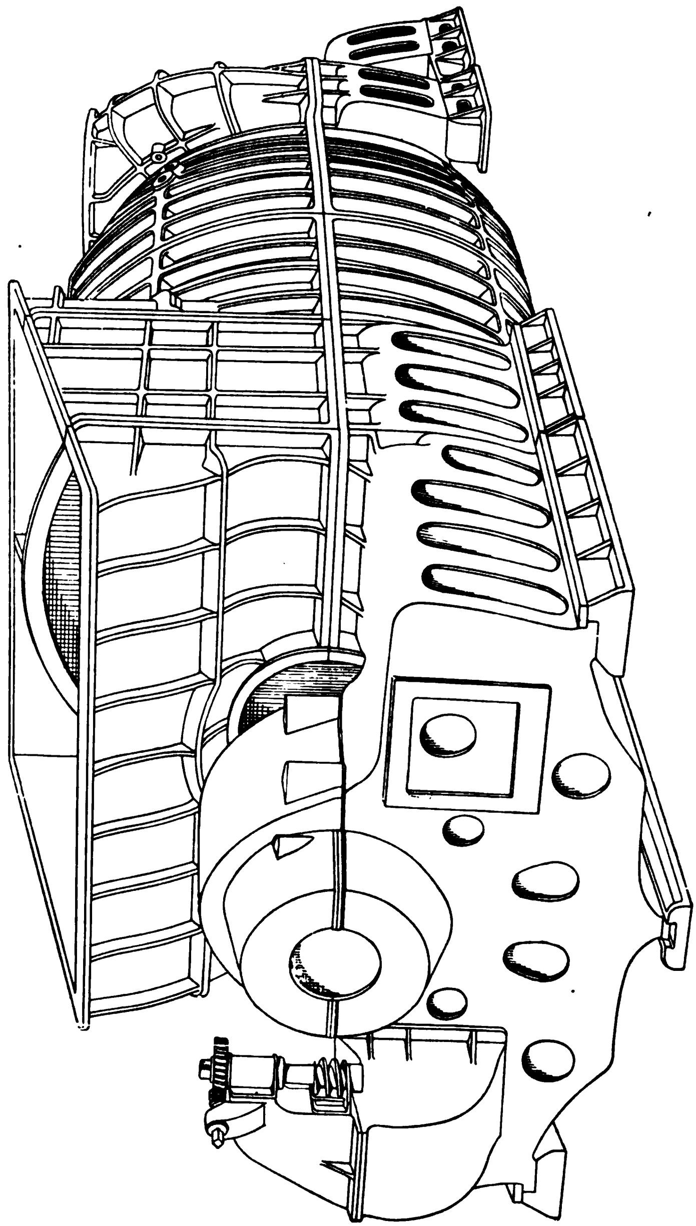

Fig. 64.

The whole casing is supported at the bearing ends by the bearing supports, which are either cast in one with the lower half or bolted to it.

The expansion of such large castings as these turbines is considerable, and it is not possible to secure them to the foundations by means of fit bolts. On the contrary, a certain amount of clearance has to be allowed in the bolt holes, and, to prevent any displacement of the casting relatively to the foundation, a sliding key is provided down the centre of the bottom of the casing. This key is free to slide in a grooved bed-plate bolted to the foundation, and so ensures a movement of the casing in line with the shaft. The arrangement of this sliding key is shown in Fig. 64.

§ 90. **Blading of the Parsons Turbine.**—The material used for the blades is usually special drawn bronze. The number of blades, dimensions, and clearance are given on the Tables No. 13 and No. 14.

Table 13.

Blading Scheme of the Parsons Turbines of a Torpedo Boat.

Arrangement.—Three shafts with one cruising turbine and one high-pressure turbine; two low-pressure turbines, and two reversing turbines;

As shown, Fig. 82, Part VII., Section I., § 121.

Maximum Output.—12,000 H.P.

Revolutions per Minute.—750.

Total Number of Blades.—Approximately 250,000.

Turbine.	Mean Blade Diameter.		No. of Rows on Drum.	Height of Blades.		Radial Clearance.		Width of Blades.				Sectional Area of Blades.			
								Minimum.		Maximum.		Minimum.		Maximum.	
	Mm.*	In.*		Mm.	In.	Mm.	In.	Mm.	In.	Mm.*	In.	Sq. mm.	Sq. in.	Sq. mm	Sq. in.
Cruising turbine - - -	700	27·5	Total 250.	18-67	·7 -2·6	1·0-1·2	·04 -·048	9	·35	13	·511	12	·018	25	·038
High-pressure main turbine -	750	29·5		50-145	1·98-5·7	1·3-1·7	·05 -·067	13	·511	16	·629	25	·038	40	·062
Low-pressure main turbine -	1,150	45·2		45-250	1·77-9·8	1·8-2·6	·07 -·1	13	·511	20	·787	25	·038	60	·093
Reversing turbine - - -	900	35·4		20-140	·79-5·5	1·9-2·7	·075-·107	13	·511	16	·629	25	·038	40	·062

* These figures are approximate only.

TABLE 14.

Blading Scheme of the Parsons Turbines of a Cruiser.

Arrangement.—Four shafts with two cruising turbines; two main high-pressure turbines; two main low-pressure turbines; two high-pressure reversing turbines; two low-pressure reversing turbines.

Arranged in two separate sets as shown, Fig. 86, Part VII., § 126.

Maximum Output.—23,000 H.P.

Revolutions per Minute.—550.

Total Number of Blades.—Approximately 650,000.

Turbine.	Mean Blade Diameter.		No. of Rows on Drum.	Height of Blades.		Radial Clearance.		Width of Blades.				Sectional Area of Blades.			
								Minimum.		Maximum.		Minimum.		Maximum.	
	Mm.*	In.*		Mm.	In.	Mm.	In.	Mm.	In.	Mm.*	In.*	Sq. mm.	Sq. in.	Sq. mm.	Sq. in.
High-pressure cruising turbine	1,150	45·2	Total 440.	20–30	·78–1·18	1·9–2·0	·075– ·079	9	·35	9	·35	12	·018	12	·018
Low-pressure cruising turbine	1,150	45·2		38–60	1·49–2·36	2·2	·0865	13	·511	13	·511	25	·038	25	·038
High-pressure main turbine	1,200	47·2		32–150	1·26–5·9	1·9–2·3	·075– ·091	13	·511	16	·629	25	·038	40	·062
Low-pressure main turbine	1,650	64·9		110–250	4·32–9·84	2·0–3·4	·079– ·134	13	·511	20	·787	25	·038	60	·093
High-pressure reversing turbine	1,150	45·2		30–75	1·18–2·95	2·2–2·6	·0865–1·02	9	·35	13	·511	12	·018	25	·038
Low-pressure reversing turbine	1,300	51·1		100–175	3·94–6·88	2·7–3·6	·107– ·142	13	·511	16	·629	25	·038	40	·062

* These figures are approximate only.

The blades are fixed in dovetailed grooves turned in the drum and have caulking pieces fitted in the grooves between adjacent blades. After a number of blades are placed in position by hand they are tapped home lightly, and after the whole row of blades is in position

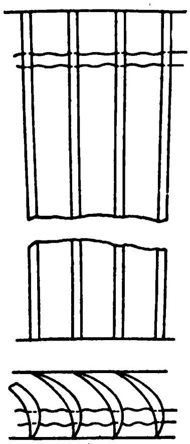

Fig. 65.

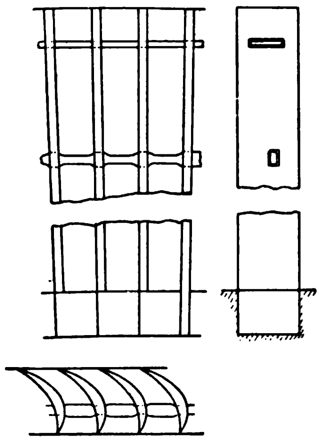

Fig. 66.

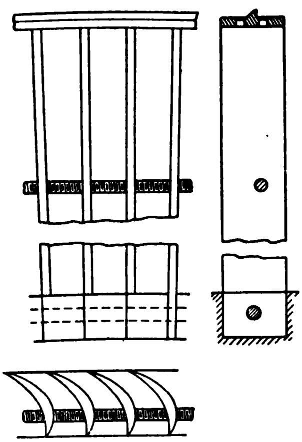

Fig. 67.

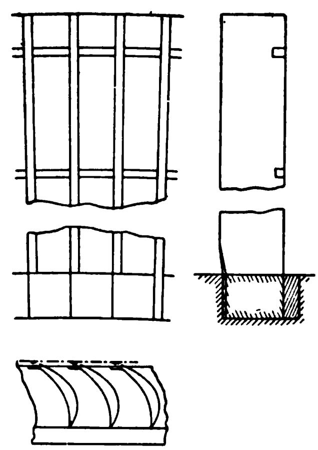

Fig. 68.

the packing pieces are caulked in by means of a special caulking tool. With blades above 1 in. in height, and especially with the long blades of the low-pressure turbines, special precautions have to be taken to ensure ample stiffness of the rows. A number of methods of doing this by means of lacing strips and shroudings, as employed in the turbine of the "Mauretania," are shown in Figs. 65 to 68. It will be noticed that the grooves for the blades in this case are not dovetailed. The various methods of binding the blades are as follows :—

Fig. 65. The blades are secured by a lacing wire passing through a hole near the top of the blade. The wire is held in position by squeezing it flat.

Fig. 66. Near the outer edge and in the centre of the blade a soft copper tube is threaded through the blades and squeezed flat between the blades.

Fig. 67. A wire is drawn through the foot of the blades and the packing pieces, and through the blades at about half their height a threaded wire, the top of the blades being shrouded. This construction enables the blading to be fixed in sections instead of individually.

Fig. 68. To secure the root of the blade a caulking strip is fitted into the groove running the whole length of the groove, and two lacing strips silver soldered in slots at the side of the blades, one near the top end and the other at about half the height.

SECTION IV.

MAIN STEAM PIPING—GLAND, DRAIN, OIL, COOLING WATER PIPING, AND LIFTING GEAR.

§ 91. **General Remarks.**—In designing the lay-out of the piping careful attention is required to obviate any damage to the turbines due to expansion of the mains, and due to water either from priming or condensation. We would refer to Bauer and Robinson, page 409 (E), for details of the design of pipe-work, valves, &c., as especially in connection with turbines is the detrimental effect on the efficiency and life of the machinery due to water so marked that special allowance must be made for large separators and efficient trapping of all water. Easy bends or expansion joints must be provided to allow of free expansion and contraction of the steam pipes, and in designing the pipe-work the considerable expansion of the turbine casing must be borne in mind.

§ 92. **Strainer.**—To prevent damage to the turbine from any solid matter carried over by the steam a strainer is fitted at the steam inlet to the turbine. This strainer can either be combined with the steam separator allowing the steam to pass through a perforated pipe inside the separator (Fig. 69), or the strainer can form a separate fitting in the main steam pipe before the steam enters the valve gear. The strainer should consist of a strong funnel-shaped sieve of non-corrosive metal (such as nickel steel) (Fig. 70) to prevent corrosion by the steam and the carrying over by the steam of any particles of the sieve.

§ 93. **Throttle Valve.**—A throttle valve forming an automatic emergency valve is fitted in the main steam pipe close to the turbine to prevent the racing of the engine in rough weather, or in case of fracture of the shaft. Fig. 71 shows such a valve of the butterfly type. The operation of the valve is controlled by a governor. As soon as the normal speed is exceeded by a predetermined amount, say 20 per cent., the governor acts on a steam relay valve admitting steam to a small cylinder *d*. Under normal working conditions the steam pressure on the piston of this cylinder keeps a pawl *c* in position, preventing the spring *a* from closing the butterfly valve. As soon as the governor on the turbine shaft causes the steam relay valve to move into the position corresponding to the 20 per cent. excess speed, steam is admitted to the underside of the steam cylinder *d*, releasing the pawl *c*, and allows the spring *a* to close the valve. A dashpot is connected to the valve lever to avoid shock on closing. To reset the valve a lever is fitted to the valve spindle and the valve opened by this

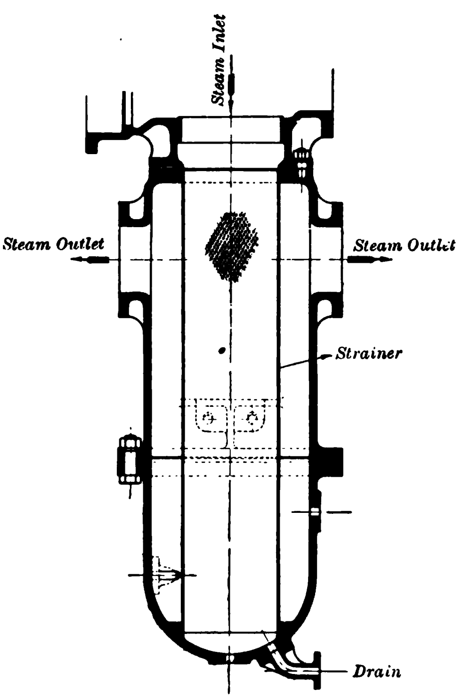

Fig. 69.

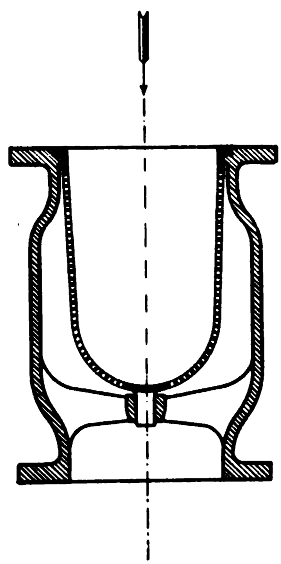
Fig. 70.

lever until the pawl *c* engages the catch controlling the opening spring. The valve will only remain in the open position if the normal or a speed lower than the normal has been attained and the relay valve is in the normal running position.

§ 94. **Reversing Gear.**—In smaller vessels such as torpedo boats, cruisers, &c., the regulating valves are balanced valves, one for ahead and one for astern motion. Frequently one or two valves are provided for cruising speeds (see Part VII., Arrangement of the Turbine).

In the single shaft arrangements of the A.E.G. and Curtis the reversing is done by closing the ahead and opening the astern valve or *vice versâ*. For the multiple shaft arrangement of Parsons (see Part VII.) special two-way valves in addition to non-return valves are required for reversing the low-pressure turbines. To ensure rapid operation the valve spindles are fitted with a steep thread so that they can be fully opened or closed by a few turns. In large installations such as the "Lusitania" and "Mauretania" two-way valves are used for reversing, these valves admitting steam according to their position either to the ahead or the astern turbines. The valves are worked by Brown steam steering engines (see Part VII.).

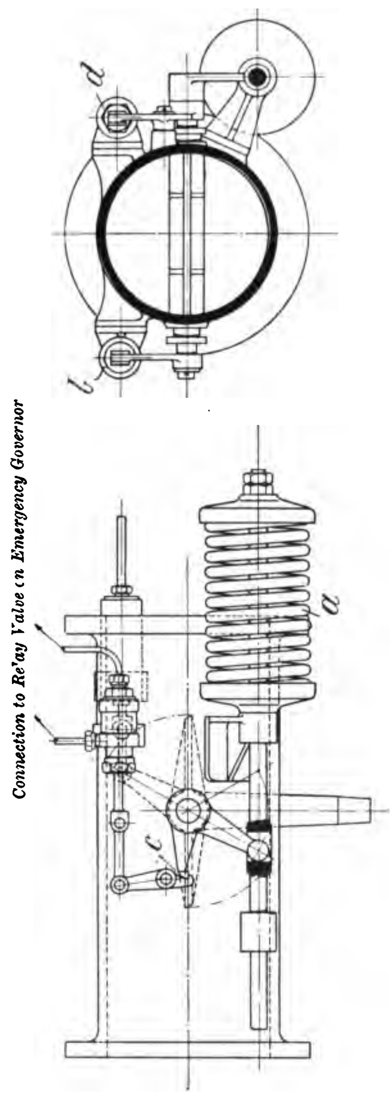

Fig. 71.

§ 95. **Glands and Drain Piping.** — The arrangement of the *Glands* and the gland *Pipe Connections* for the A.E.G. and the Parsons turbines have been described in § 75 and § 88. The description of the glands given above also deals with the necessary pipe connections.

Drain Piping.—It is necessary to make provision for careful drainage of the turbines, and especially of the low-pressure cylinder, as under no conditions must the blading work under water. If at all possible the wet air pump should be mounted sufficiently low to allow the condensed steam from the low-pressure cylinder to flow to the air pump with a head of 200 to 300 mm. (8 to 12 in.).

With a smaller head than this the water pressure may not be sufficient to lift the suction valves of the air pump, so that the turbine is apt to become water logged. Where drainage to the air pump is not possible the drain pipe is taken to a separate drain trap placed in the bottom of the ship. After closing a valve in the drain pipe the condensed water is discharged from the trap by boiler steam to the condenser. Instead of using boiler steam direct a steam ejector can be used, but in this case the drain water requires to be cooled before being ejected, as otherwise the ejector is apt to fail.

The draining of the high-pressure turbine is much simpler, as the steam pressure is sufficiently high in these stages to discharge the water direct into the condenser. A suitable number of drain cocks are provided on the high-pressure cylinder. The drain water leaving the high-pressure turbine under pressure can also be utilised for discharging the contents of the above-mentioned drain traps of the low-pressure cylinder to the condenser.

All drain valves are operated by levers from the starting platform, and care should be taken in designing to arrange them in a handy and systematical fashion.

§ 96. **Oil Piping.**—Instead of using wick lubrication for the main and thrust bearings as in reciprocating engines the oil is supplied under pressure. The oil is taken from a tank and pumped by a duplex or wheel pump under a pressure of 2 to 4 atm. (30 to 60 lbs. per sq. in.) to the bearings. As only the bottom half of the main bearings is loaded the oil is delivered to the bottom half and distributed over the shaft through oil grooves (see Fig. 42). The oil is collected in the bearing pedestal from where it flows under a natural head through large pipes back to the oil tank. This tank must have a drain connection for running off any water which may be carried over with the oil from the glands, &c.

With this method of forced lubrication where oil at a high pressure is supplied to the bearing there is no actual contact between the shaft and the bearing. Wear of the bearing surfaces is, therefore, reduced to a minimum, and as large quantities of oil are pumped through the bearings it maintains its lubricating qualities for long periods. A further advantage is the cooling effect on the bearings of the oil itself. The oil after returning to the oil tanks and passing through the pumps is forced through an oil cooler before being delivered to the bearings.

The oil cooler is a closed vessel connected to the circulating water mains, the oil passing through a coil of tubes inside the cooler.

Oil under pressure is also supplied to the thrust block.

§ 97. **Cooling Water Piping.**—With forced lubrication direct cooling of the bearings is impossible. For this reason the castings of the bearings themselves or the lower castings of the bearing pedestals are hollow and connected to the cooling water mains.

The cooling water is taken from the discharge side of the circulating water pump and supplied to a main cooler pipe line from which branch pipes are led to the oil cooler and bearings. The water after passing through these is generally taken by a common main leading to the suction side of the circulating water pump.

The head between the suction and discharge side of the pump is generally sufficient to supply the requisite amount of water to the bearings and cooler.

§ 98. **Lifting Gear.**—Suitable tackle is necessary to raise the cylinder covers as well as the rotors for inspection. This gear must be arranged to prevent any fouling of the blades when lifting, and allow of the parts being raised parallel to their normal position.

In the smaller Parsons turbine strong guide bolts are screwed into the lower flange of the cylinder, two at each end, these serving to guide the flange of the cylinder during lifting. Chain blocks running on beams fixed under deck above the turbine serve to lift the cover.

If the rotor has to be lifted, the weight of the casing is taken up by distance pieces between the flanges, and the rotor lifted by the blocks. For doing this, either steel bands lined with leather, copper, or white metal are passed over the ends of the shafts, or a lifting bar with two hooks going round the shaft or inside the drum is used.

Before lifting, all pipe connections, fittings, &c., are removed from the top covers.

For lifting the covers of large turbines lifting screws take the place of the chain blocks. These screws pass through a cross-bar which moves along the guide bolts. To raise the rotor the steel bands or hooks are bolted to the cross-bar.

The lifting gear of the A.E.G. turbine is of this type. The guide rods rest on the bearing blocks and are bolted at their upper end to the deck. The cover is guided by heavy lugs bolted to the casing end covers in place of the nozzle box.

A cross-bar passing under the guide lugs serves to lift the cover.

The actual lifting gear is fitted to a beam supported by the guide rods above the turbine (Fig. 72). A chain drum at either end of the beam for lifting both cross-bars is driven by chain wheel, worm and worm-wheel.

After supporting the cylinder cover and again lowering the cross-bar

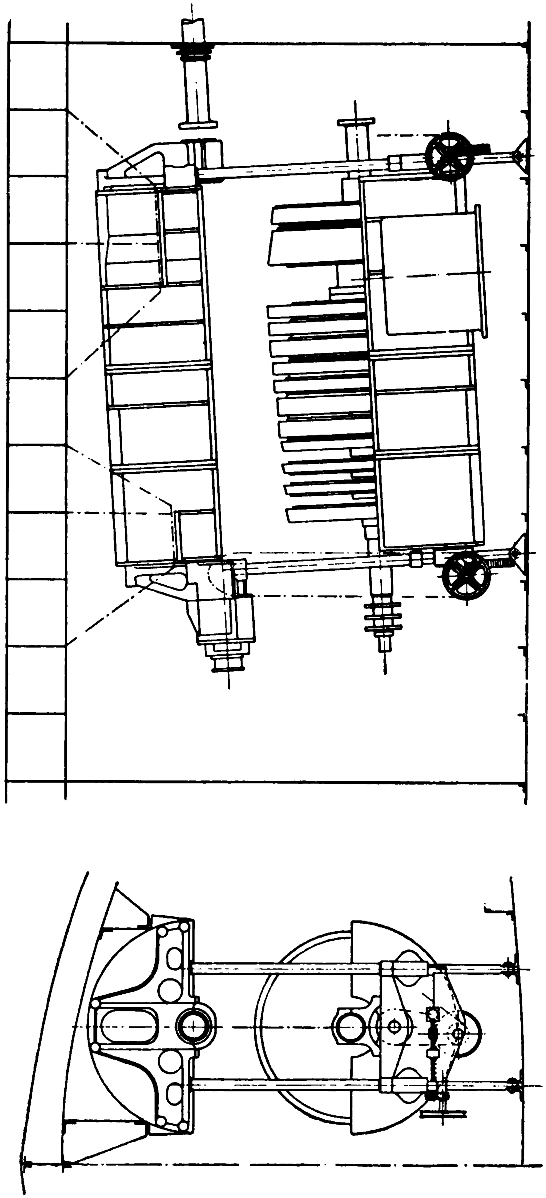

Fig. 72.

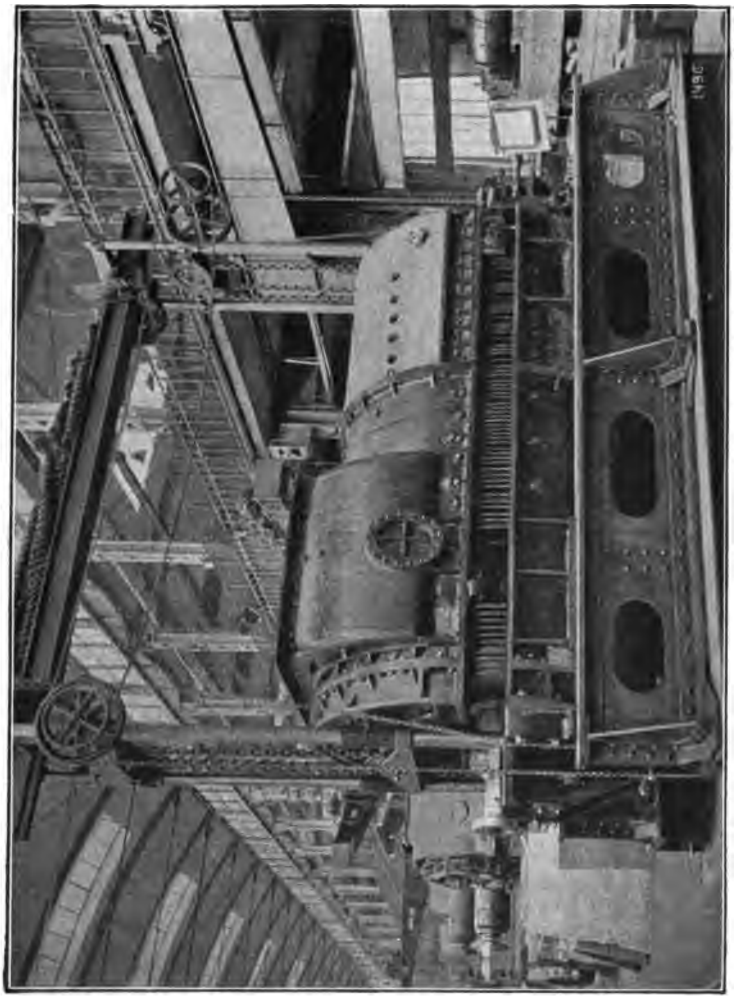

Fig. 73.

the rotor can be raised by means of the hooks or steel bands which are bolted to the cross-bar.

To ensure both ends being raised parallel the two lifting gears are worked by one and the same chain wheel.

Fig. 73 shows the lifting gear of a marine steam turbine in the Berlin works of the A.E.G. in the act of raising the top cover of a turbine for inspection.

PART V.

SHAFTING AND PROPELLERS.

SECTION I.

SHAFTING.

§ 99. **Propeller Thrust.**—The method of determining the axial thrust is the same for turbine steamers as for steamers fitted with reciprocating engines (see Bauer and Robertson, "Marine Engines," page 335). In the case of reciprocating engines all calculations for thrust are made with reference to the indicated horse-power. A similar method is adopted for turbine steamers with the difference that the calculations are based on the effective horse-power of the turbine. If S.H.P. is the effective or shaft horse-power of the turbine, n the number of revolutions per minute, H the pitch of the propeller in meters (or feet), P the thrust in kg. (or lbs.), then the equation will be :—

$$P = \frac{\text{S.H.P.} \cdot 4{,}500}{n \cdot H} \text{ in kg.,}$$

or

$$P = \frac{\text{S.H.P.} \cdot 33{,}000}{n \cdot H} \text{ in lbs.}$$

The effective thrust is the above value of P multiplied by the efficiency of the propeller, and by substituting $n \cdot H$ by the speed of the vessel (see § 53). In the following the dimensions of propeller are, for the sake of simplicity, calculated from the effective thrust P as calculated above.

§ 100. **Torque.**—The torque transmitted by the turbine to the main shaft is :—

$$M = 71{,}620 \frac{\text{S.H.P.}}{n} \text{ in cm. kg.,}$$

or,

$$M = 63{,}000 \frac{\text{S.H.P.}}{n} \text{ in inch lbs.}$$

The number of revolutions of turbine steamers is considerably higher than in steamers with reciprocating engines, and consequently the torque is less in the former, assuming the same output. The shafting of the turbine steamers can therefore be made comparatively light.

The advantage is further increased by the fact that the turbine exerts an even turning moment, whereas the variations in the torsional stresses in the shafting of a reciprocating steamer may be considerable (see Bauer and Robertson, "Marine Engines," § 34).

§ 101. **Material of the Shafts.**—For this reason considerably higher stresses are allowable than in reciprocating steamers.

The torsional stress is usually as follows:—

$s = 420$ to 550 kg. per cm.^{2} in cargo and passenger steamers,
$s = 500$ to 650 „ in battleships and cruisers, and
$s = 750$ to 850 „ in torpedo boats and destroyers;

or in English measures:—

$s = 5{,}970$ to $7{,}820$ lbs. per sq. in. in cargo and passenger steamers,
$s = 7{,}110$ to $9{,}240$ „ in battleships and cruisers,
$s = 10{,}680$ to $12{,}100$ „ in torpedo boats and destroyers.

The material used for the shafts is generally Siemens-Martin steel of a breaking stress of 4,500 to 5,300 kg. cm.^{2} (64,000 to 75,000 lbs. per sq. in.), in torpedo boats of 5,000 to 6,400 kg. cm.^{2} (71,000 to 91,000 lbs. per sq. in.), and an elongation of 18 to 20 per cent., the lower value of the elongation being for the higher stress.

In view of the bending moment it is an advantage to allow for a large moment of inertia of the shaft, and for this reason they are frequently made hollow (this is nearly always the case in warships). It is usual to make the inside diameter about ·4 to ·65 of the outside diameter.

§ 102. **Shaft Couplings.**—The couplings, flanges, and bolts are of the same types and dimensions as those employed with reciprocating engines (see Bauer and Robertson, "Marine Engines," page 343).

§ 103. **Plummer Blocks or Bearings.**—The distance of the bearings apart varies according to the diameter of the shaft, and is generally 3 to 6 m. or 9 to 18 ft. In view of the high velocity of the shaft, large and rigid bearings must be provided.

Length of bearings is approximately 1·25 to 1·4 times the shaft diameter for small vessels.

„ „ approximately 1·0 to 1·2 times the shaft diameter for large vessels.

The bearing consists of cast iron lined with white metal in both halves.

The thickness of the white metal lining is

$$\delta = \text{about } \cdot01\, d + 5 \text{ mm. in small, and}$$
$$\delta = \text{about } \cdot01\, d + 9 \text{ mm. in large bearings;}$$

or in English units:—

$$\delta = \text{about } \cdot01\, d + \tfrac{3}{16} \text{ in. in small, and}$$
$$\delta = \text{about } \cdot01\, d + \tfrac{3}{8} \text{ in. in large bearings,}$$

d being the shaft diameter in mm. or inches.

The plummer blocks are of cast iron or cast steel (see Fig. 74). The large bearings have separate sleeves of cast steel or brass with white metal lining, which in the case of the smaller sizes is cast direct into the bearings.

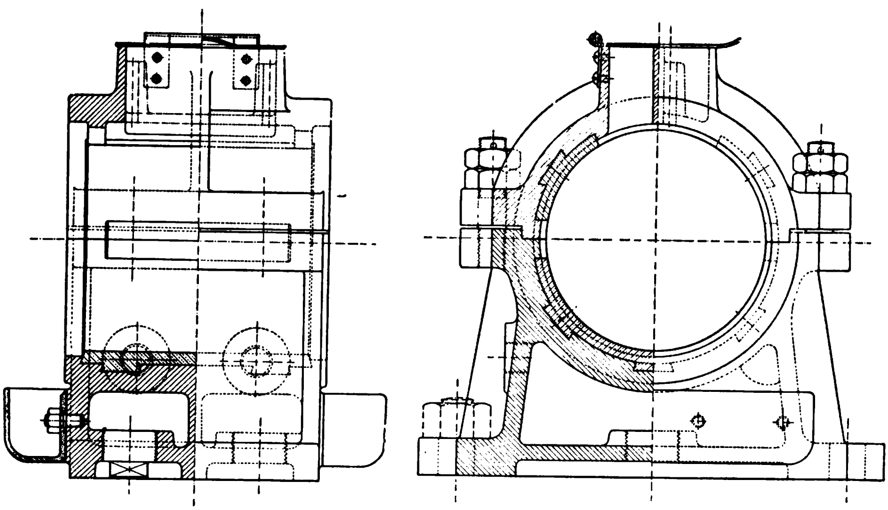

Fig. 74.

Ample lubrication and cooling of the bearings is essential. Wick lubrication, and in some cases forced lubrication, is used. For the cooling either direct cooling in the usual way by a connection from the circulating pump is used, and in addition the lower part is generally indirectly cooled.

If the same oil pumps are used for supplying oil to both the turbine bearings and the plummer blocks, care must be taken that the oil supply to the highly loaded turbine bearings is not interfered with by the oil taken by the plummer blocks, and the oil pipes to the latter should be of small size and the quantity adjustable by cocks placed at each bearing. In some cases ring lubricated bearings for the plummer blocks can be used.

§ 104. **Tail or Propeller Shaft.**—The diameter of this is generally about 3 to 6 per cent. greater than that of the main shaft. In the best designs the tail shaft has a sheathing in the bearings of nickel steel (containing about 20 to 25 per cent. nickel) as a protection against corrosion. The thickness of this sheath is 15 to 20 mm. ($\frac{5}{8}$ to $\frac{3}{4}$ in.) in large, and 7 to 10 mm. ($\frac{1}{4}$ to $\frac{3}{8}$ in.) in small and fast vessels. The sheath projects beyond the bearing by about 40 to 100 mm. ($1\frac{1}{2}$ to 4 in.) to enable unequal lining up to be adjusted.

For installations in the mercantile marine where a cheaper plant is permissible the nickel steel sheathing is often dispensed with, but the diameter of the shaft made slightly larger in the bearings. In these designs means of amply lubricating the shaft in the stern tube are provided (see § 105).

The bore of the propeller shaft is usually the same as that of the main shafts. To prevent too great a loading of the shaft in the bearings the diameter of the bore of the after part of the shaft is reduced by one-half. The material of the shaft is the same as given in § 101. Inside the stern tube the shaft is in contact with sea water, and is protected against any corrosive action by a sheath of rubber 5 to 10 mm. ($\frac{3}{16}$ to $\frac{7}{16}$ in.) thick.

Ample bearing surface is provided, based on the following data:—

In the plummer block, 4 to 4·5 times the shaft diameter.

At the after stern tube bearing, 4·2 to 5·2 times the shaft diameter.

At the forward stern tube bearing, 2·3 to 3·0 times the shaft diameter.

Special attention must be given to the support of the shafting to ensure that the critical speed of the shaft does not approach the maximum working speed (see § 46).

The propeller shaft is generally mounted so that it can be withdrawn from the stern, and for this purpose the diameter of the journals are reduced by 2 to 5 mm. ($\frac{3}{64}$ to $\frac{3}{16}$ in.) as compared to the next preceding one. In quite large vessels the shaft is frequently arranged that it can be withdrawn toward the front or in either direction. In the latter case the bearings are all made of the same diameter. The taper of the propeller cone is 1 : 10 or 1 : 12.

§ 105. **Stern Tube and Plummer Block.**—The stern tube is built up in the same manner as for ordinary marine engines (Bauer and

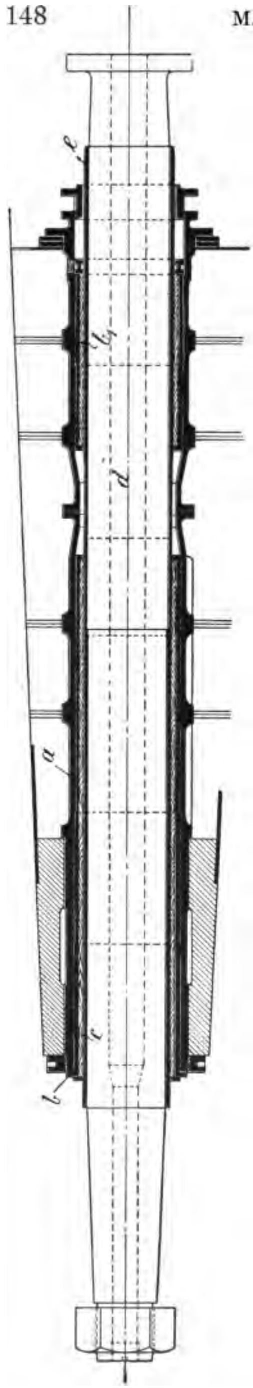

Fig. 75.

Robertson, "Marine Engines," page 346). In the mercantile marine this is usually of cast iron and a brass casting in large naval vessels. In torpedo boats they are usually made of forged steel. The thickness of the latter is 6 to 8 mm. ($\frac{1}{4}$ to $\frac{5}{16}$ in.). If the length exceeds 16 or 20 diameters (see Fig. 75), that is about 7 m. (23 ft.), the tube is made in two lengths. The stern tube is sometimes fitted in from the stern to allow it to be withdrawn with ease when the ship is in dry dock. The after stern tube bush in that case must be made 2 mm. larger in diameter than the largest stern tube diameter. In small vessels the two bearings are usually lined with white metal, and in large vessels (see Fig. 75) are lined with lignum vitæ. The thickness of the lignum vitæ bearing is $c = 20$ to 30 mm. ($\frac{13}{16}$ to $1\frac{3}{16}$ in.). The shaft when newly fitted should have a clearance of 1 to 2 mm. ($\frac{1}{32}$ to $\frac{3}{64}$ in.) in the bearing so as to allow for expansion of the wood under water. The lignum vitæ is fitted in bronze bushes b, b^1 drawn into the stern tube a. If lignum vitæ is used for the bearing the shaft is fitted with a brass sleeve e, 15 to 25 mm. ($\frac{5}{8}$ to 1 in.) thick, dependent on the thickness of the shaft.

White metal bearings have a thickness of about 12 to 15 mm. ($\frac{1}{2}$ to $\frac{5}{8}$ in.). In a length of 200 mm. (8 in.) the outside of the bush is stepped by 1 to 2 mm. ($\frac{1}{32}$ to $\frac{3}{64}$ in.) to facilitate fitting.

In vessels with separate plummer blocks the bush is of cast steel lined with white metal, and the bush is arranged so that it can be withdrawn from the stern. To allow access of the sea water and oil to the bearing, grooves are machined in the white metal, the number of these being ten to sixteen, about 18 to 25 mm. ($\frac{11}{16}$ to 1 in.) wide, and 3 to 6 mm. ($\frac{1}{8}$ to $\frac{1}{4}$ in.) deep. To damp any high frequency vibrations which may occur

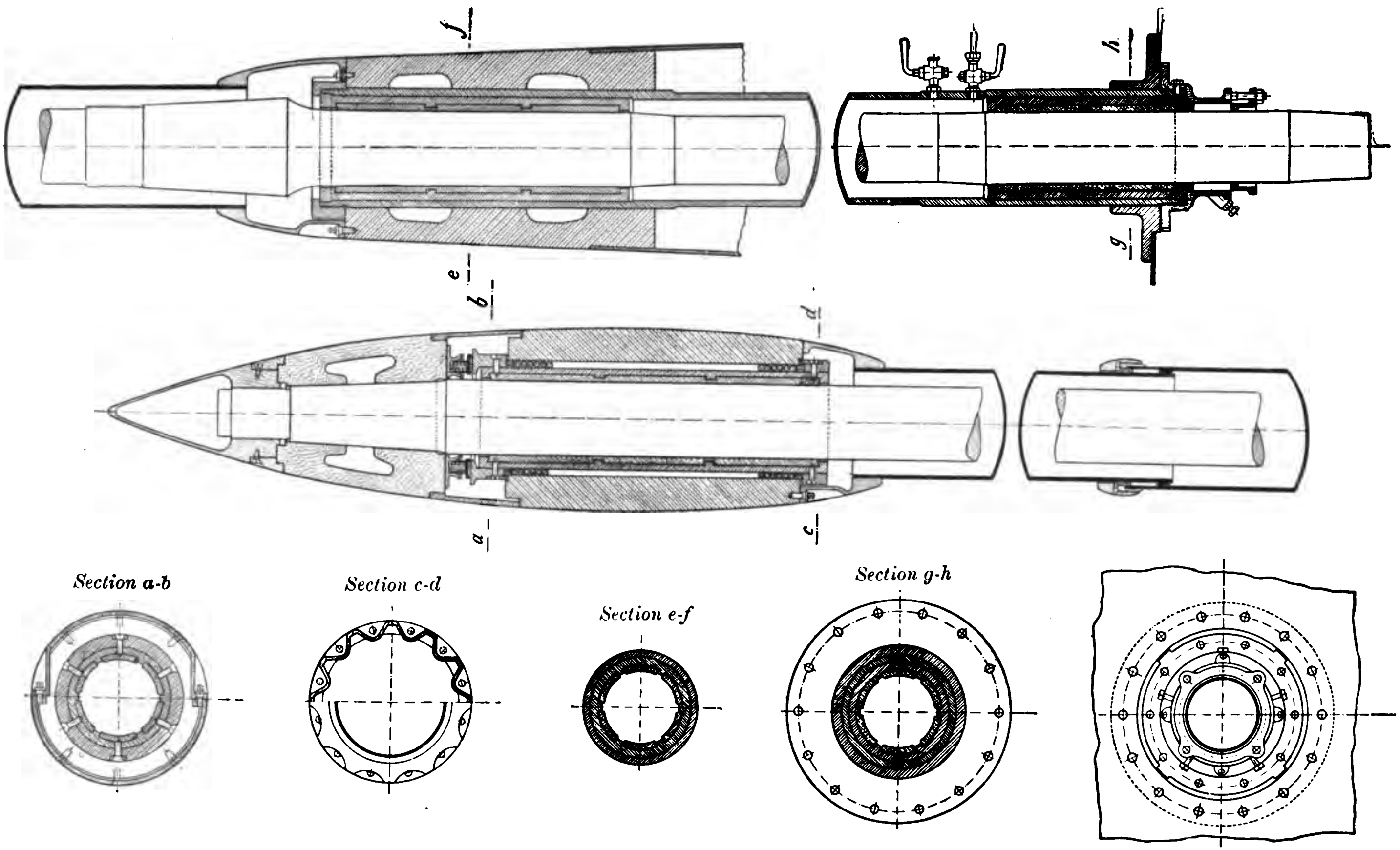

Fig. 76.

11

at the stern strong rings used to be inserted between the steel bush and shaft, though this complication is rarely resorted to now.

The forward end of the stern tube is fitted with a stuffing box of ample dimensions.

Width of stuffing box:—

$a = \cdot 06d + 6$ mm., or $\cdot 06d + \frac{1}{4}$ in. for torpedo boats,
$a = \cdot 08d + 8$ mm., or $\cdot 08d + \frac{5}{16}$ in. for larger vessels.

Depth of packing space:—

$t = 4$ to $5a$ for torpedo boats,
$t = 5$ to $6{\cdot}5a$ for larger vessels.

The stuffing-box nuts are provided with a toothed rim to screw them on evenly (Bauer and Robertson, "Marine Engines," page 352).

Oil under pressure is sometimes admitted to the stuffing box.

To avoid loss of oil the mercantile marine stuffing-box arrangement of the Cederwell-Toussaint type is sometimes used (see Fig. 76).

It is now usual to enclose the propeller shaft in a tube between the stern tube where the shaft comes through the skin of the vessel and the bearing bracket to reduce the resistance at the stern of the vessel.

SECTION II.

PROPELLER.

§ 106. **General Remarks.**—The formula used for calculating ordinary propellers can be used for calculating the high-speed propellers in use on turbine-driven vessels (see Bauer and Robertson, "Marine Engines," pages 361 and ff.), but different values must be given to the constants K_1 and K_2. The following method of calculation has, however, proved more suitable.

§ 107. **Pitch and Diameter.**—In a screw moving in a fixed nut the efficiency decreases with decreasing pitch, and in the same manner the efficiency of a marine propeller decreases if the ratio pitch to diameter drops below a certain value (about unity).

This is the reason for the difficulty in designing a suitable turbine propeller. Due to the high number of revolutions, a small pitch is necessary, whereas a large diameter is required to obtain a low specific pressure on the blades of the propeller.

§ 108. **Propeller Velocity.**—Tests made originally by the Hon. C. A. Parsons, and later at the "Vulcan" Works, Stettin, have demonstrated that a high propeller velocity reduces the efficiency of the propeller, leading to the necessity of obtaining a suitable ratio of pitch to diameter by choosing a large pitch and reducing the diameter.

Between the ratio of $\frac{H}{D}$ and the blade tip velocity, the following formula holds good :—

$$\frac{H}{D} = \text{constant} \times \frac{v}{(1-s)u}.$$

Where v is the speed of the vessel,
s is the slip, and
u is the tip velocity of the propeller blade.

This formula indicates that the choice of $\frac{H}{D}$ is limited by u.

§ 109. **Surface Pressure.**—Keeping in mind that the ratio of pitch to diameter, as well as the velocity u and the thrust per unit of area of the propeller, must not exceed a maximum to obtain an efficient propeller, Parsons in designing the propellers of the "Turbinia" arranged a number of propellers on a single shaft. This enables a large blade surface to be obtained, and at the same time to maintain a small propeller diameter and low propeller velocity. The experiment with the "Turbinia" was perfectly successful, for in spite of the high number of revolutions no "cavitation" effect resulted. Later this method was abandoned, due to the complication of the stern frame of the vessel, and the consequent increase of resistance of the vessel in the water. Fig. 77 shows the original arrangement of the four shafts of a cruiser, each fitted with two propellers. At the present only one propeller per shaft would be used. In order to reduce the pressure on the propeller, a large propeller area would appear advisable, this being obtained by having a high value of the ratio:

$$a = \frac{\text{projected propeller area}}{\text{circular area}}.$$

Unfortunately, experience has shown that it is not advisable to make this ratio greater than $a = \cdot 65$ (see Table 15 below).

§ 110. **Slip.**—The slip increases with decreasing blade area, and decreasing circumferential velocity. That the latter is the case is obvious. The former can be explained by the fact that an increase in the velocity leads to an increase in the resistance exerted by the water on the blades. Even this increase in resistance only holds good up to the limits at which continuity of flow of the water exists.

For turbine propellers it is usual to work with a slip of about 20 per cent. at full speed and power. This varies, however, with the type of vessel, and the figures usually chosen are:—

Torpedo boats - - - -	$s = 18$ to 25 per cent.
Cruisers - - - - -	$s = 12$ to 22 ,,
Passenger boats - - -	$s = 10$ to 20 ,,

Higher or lower values are sometimes chosen, but it is usual for the slip to be inside these limits.

§ 111. **Maximum Number of Revolutions.**—Experience has shown that to obtain a workable propeller efficiency a certain limiting value for the ratio pitch to diameter, specific surface pressure, tip velocity and ratio of the projected to the circular area must not be exceeded or reduced as the case may be. A reduction of the number of revolutions of the propeller favourably influences all the above factors, and it will be the designer's object to choose such a maximum

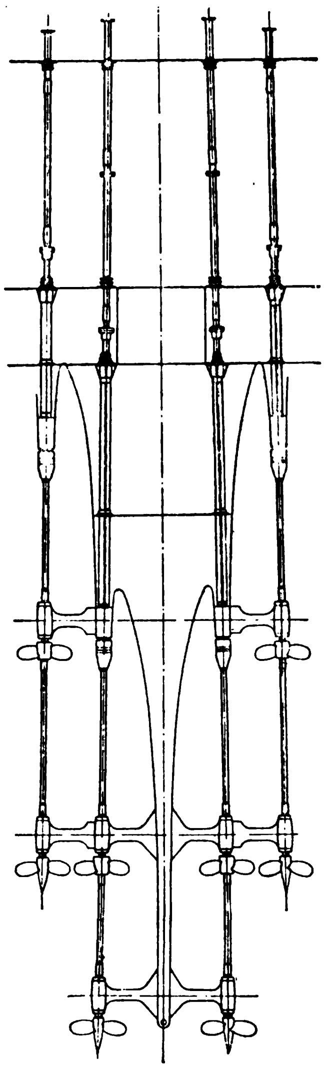

Fig. 77.

speed that the value of the above-mentioned factor remains within suitable limit.

The following table gives these values for different classes of vessels :—

TABLE 15.

Limiting Values for Propeller Design of Turbine-Propelled Vessels.

Type of Vessel.	Peripheral Velocity u of the Blade Tips. Maximum Values.		Ratio of Pitch to Diameter, $\frac{H}{D}$.	Ratio of Projected Surface to Circular Area of Blades "a." Maximum Values.	Thrust on the Projected Surface from Formula in § 99. Maximum Values.	
	M per sec.	Feet per sec.*			Kg. per sq. m.	Lbs. per sq. ft.*
1. Torpedo boats of 30 knots and above	65 to 75	213 to 246	·95 to 1·10	·54 to ·61	1·23 to 1·57	·252 to ·322
2. Torpedo boats of 25 to 39 knots	55 to 65	180 to 213	·92 to ·98	·52 to ·58	1·12 to 1·40	·23 to ·287
3. Small cruisers of 22 to 26 knots	52 to 58	170 to 190	·85 to ·95	·52 to ·58	1·06 to 1·40	·217 to ·287
4. Large cruisers of 20 to 24 knots	50 to 53	164 to 173	·85 to ·92	·52 to ·58	1·03 to 1·12	·211 to ·23
5. Battleships of 19 to 22 knots	48 to 52	157 to 170	·80 to ·90	·52 to ·58	1·00 to 1·10	·205 to ·225
6. Passenger boats of 18 to 21 knots	45 to 50	147 to 164	·75 to ·85	·50 to ·56	·93 to 1·00	·191 to ·205

* In round figures.

This table does not give any values for vessels for a lower speed than 18 knots.

At the present position of the art the use of turbines for the direct propulsion of vessels at a lower speed than 18 knots does not appear to be economical, and the above table therefore does not give any values for a lower speed than this.

The table shows that with increasing speed of the vessel the number of revolutions can be increased similar to vessels fitted with reciprocating engines where for high-speed vessels a higher engine speed is employed than with low-speed vessels.

§ 112. **Example for the Calculation of the Permissible Number of Revolutions.**—The following example will show the calculations necessary to determine the maximum permissible number of revolutions for a given vessel.

Assuming that the vessel is a torpedo boat designed for a speed of 33 knots driven by two 4,500 H.P. steam turbines, Table 15 gives the highest peripheral velocity as $u = 75$ m. (246 ft.).

If a value of $u = 72$ m. (236 ft.) is assumed, then the diameter will be:

$$\mathrm{D} = \frac{72 \,.\, 60}{\pi \,.\, n} \text{ metric units,}$$

$$\mathrm{D} = \frac{236 \,.\, 60}{\pi \,.\, n} \text{ English units.}$$

In this formula both D and n are unknown quantities, and to enable a preliminary result to be obtained the simplest course is to assume the number of revolutions. If these are:

$$n = 800,$$

then

$$\mathrm{D} = \frac{72 \,.\, 60}{3{\cdot}14 \,.\, 800} = 1{\cdot}72 \text{ m.,}$$

or,

$$\mathrm{D} = \frac{236 \,.\, 60}{3{\cdot}14 \,.\, 800} = 5{\cdot}62 \text{ ft.}$$

The pitch H can be calculated from the following formula, in which s denotes the slip:—

$$\frac{n \,.\, \mathrm{H} \,.\, 60}{1{,}852} = \frac{v}{1 - s} \text{ metric units,}$$

$$\frac{n \,.\, \mathrm{H} \,.\, 60}{6{,}080} = \frac{v}{1 - s} \text{ English units.}$$

If

$$s = 20 \text{ per cent.},$$

then

$$1 - s = \cdot 8,$$

and

$$\text{H} = \frac{33}{\cdot 8} \cdot \frac{1{,}852}{800 \,.\, 60} = 1 \cdot 59 \text{ m. metric units,}$$

$$\text{H} = \frac{33}{\cdot 8} \cdot \frac{6{,}080}{800 \,.\, 60} = 5 \cdot 21 \text{ ft. English units,}$$

and consequently

$$\frac{\text{H}}{\text{D}} = \frac{1 \cdot 59}{1 \cdot 72} = \cdot 925 \text{ metric units,}$$

$$\frac{\text{H}}{\text{D}} = \frac{5 \cdot 21}{5 \cdot 62} = \cdot 925 \text{ English units.}$$

From Table 15 this value is somewhat low, but before going into this any further, the pressure per unit of surface may be calculated.

The surface is $\frac{\text{D}^2 \pi}{4} = 2 \cdot 32$ sq. m., or 25 sq. ft.

The calculated thrust (see § 99) is:

$$\text{P} = \frac{\text{S.H.P.} 4{,}500}{n \,.\, \text{H}} = \frac{4{,}500 \,.\, 4{,}500}{800 \,.\, 1 \cdot 59} = 15{,}900 \text{ kg. metric measures,}$$

or, $$\text{P} = \frac{\text{S.H.P.} 33{,}000}{n \,.\, \text{H}} = \frac{4{,}500 \,.\, 33{,}000}{800 \,.\, 5 \cdot 21} = 35{,}100 \text{ lbs. English measures.}$$

Let $a = \cdot 56$ be the ratio of the projected area to the area of the propeller circle, then the projected area will be:

$$\text{A} = 2 \cdot 32 \,.\, \cdot 56 = 1 \cdot 30 \text{ sq. m., or 14 sq. ft.,}$$

and the thrust per unit of blade surface,

$$p = \frac{15{,}900}{13{,}000} = 1 \cdot 22 \text{ kg. per sq. cm. metric units,}$$

$$p = \frac{35{,}100}{14 \,.\, 144} = 17 \cdot 4 \text{ lbs. per sq. in. English units,}$$

a value which lies within the permissible limit.

All values now lie within the limits, but the ratio of pitch to diameter is somewhat low, and consequently the diameter too large. Altering the number of revolutions would not be an improvement, as D and H would be varied in proportion, and the ratio $\frac{\text{H}}{\text{D}}$ would remain constant. Consequently, the peripheral velocity will have to be reduced. If this is made

$$u = 70 \text{ m., or } 230 \text{ ft.,}$$

then,

$$D = 1{\cdot}67 \text{ m., or } 5{\cdot}5 \text{ ft.,}$$

and

$$\frac{H}{D} = \frac{1{\cdot}59}{1{\cdot}67} = {\cdot}95, \text{ or } \frac{H}{D} = \frac{5{\cdot}21}{5{\cdot}5} = {\cdot}95,$$

a value which is within the limits.

The projected area becomes:

$$\frac{1{\cdot}67^2\pi}{4} \times {\cdot}56 = 1{\cdot}23 \text{ sq. m., or } 13{\cdot}25 \text{ sq. ft.,}$$

and the thrust per unit of propeller surface,

$$p = \frac{15{,}400}{12{,}300} = 1{\cdot}29 \text{ kg. per sq. cm., or } 18{\cdot}3 \text{ lbs. per sq. in.}$$

It is not advisable to assume a higher value when designing the plant, for the guaranteed output of the turbines is mostly over-estimated, and the thrust becomes greater than originally estimated.

As all the calculated data fall within the values given in Table 15, they can be accepted for the design, showing that the chosen number of revolutions was correct.

§ 113. **Calculation of the Permissible Number of Revolutions from Table 15.**—A trial and error method was adopted in the preceding paragraph to obtain the value of the number of revolutions, but it is also possible to calculate these from the values given in Table 15.

The peripheral velocity is:—

$$u = \frac{D\pi \, . \, n}{60}, \text{ or } D = \frac{u \, . \, 60}{\pi n}.$$

Further,

$$\frac{n \, . \, H \, . \, 60}{1{,}852} = \frac{v}{1-s} \text{ metric units,}$$

$$\frac{n \, . \, H \, . \, 60}{6{,}080} = \frac{v}{1-s} \text{ English units.}$$

From these formulæ:

$$\frac{H}{D} = \frac{1{,}852 \, . \, \pi \, . \, v}{3{,}600 \, . \, (1-s)u} \text{ metric units,}$$

$$\frac{H}{D} = \frac{6{,}080 \, . \, \pi \, . \, v}{3{,}600 \, . \, (1-s)u} \text{ English units.}$$

If we denote $\frac{v}{1-s}$ with C, then the above can be written:

$$\frac{H}{D} = 1{\cdot}615\frac{C}{u} \text{ metric units,}$$

$$\frac{H}{D} = 5{\cdot}3\frac{C}{u} \text{ English units.}$$

This formula shows that the peripheral velocity and the ratio pitch to diameter depend upon one another; so that if for a given speed of the vessel v, and a given slip s, the ratio $\frac{H}{D}$ is assumed, the velocity u of the propeller is determined.

The thrust is:—

$$P = \frac{4{,}500 \,.\, \text{S.H.P.}}{n \,.\, H} \text{ kg. metric units,}$$

$$P = \frac{33{,}000 \,.\, \text{S.H.P.}}{n \,.\, H} \text{ lbs. English units,}$$

and the projected propeller blade area:

$$A = \frac{D^2\pi}{4} \,.\, a.$$

From this the pressure per unit of blade surface is:

$$p = \frac{P}{A} = \frac{4 \cdot 4{,}500 \,.\, \text{S.H.P.}}{n \,.\, H \,.\, D^2 \,.\, \pi \,.\, a} = 5{,}725 \frac{\text{S.H.P.}}{n \,.\, H \,.\, D^2 \,.\, a} \text{ kg. per sq. m.,}$$

and $$n = 5{,}725 \frac{\text{S.H.P.}}{p \,.\, H \,.\, D^2 a},$$

or $$p = \frac{P}{A} = \frac{4 \,.\, 33{,}000 \text{ S.H.P.}}{n \,.\, H \,.\, D^2 \pi a} = 42{,}000 \frac{\text{S.H.P.}}{n \,.\, H \,.\, D^2 a} \text{ lbs. per sq. ft.}$$

and $$n = 42{,}000 \frac{\text{S.H.P.}}{p \,.\, H \,.\, D^2 a}.$$

Now, $$D^2 = \frac{u^2 \,.\, 60^2}{3{\cdot}14^2 \,.\, n^2},$$

and $$H = \frac{v}{(1-s)} \cdot \frac{1{,}852}{n \,.\, 60} \text{ metric units,}$$

$$H = \frac{v}{(1-s)} \cdot \frac{6{,}080}{n \,.\, 60} \text{ English units.}$$

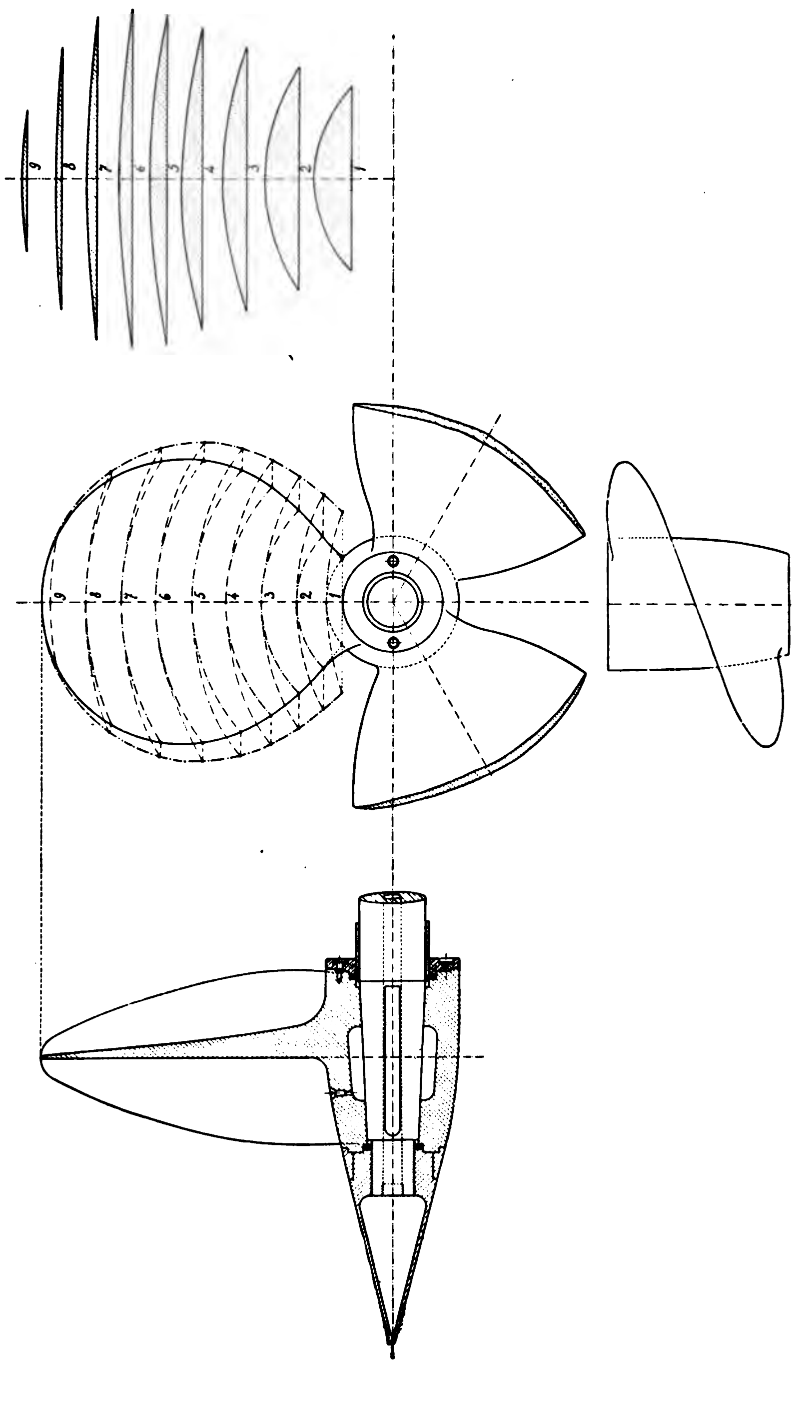

Fig. 78.

Inserting these values in the above formula, we obtain the following:

$$n = 5{,}725 \frac{\text{S.H.P.}}{p \, . \, a} \cdot \frac{\pi^2 \, . \, n^2 \, . \, (1 - s)n \, . \, 60}{u^2 \, . \, 60^2 \, . \, v \, . \, 1{,}852} \text{ metric units,}$$

$$n = 42{,}000 \frac{\text{S.H.P.}}{p \, . \, a} \cdot \frac{\pi^2 \, . \, n^2 \, . \, (1 - s)n \, . \, 60}{u^2 \, . \, 60^2 \, . \, v \, . \, 6{,}080} \text{ English units,}$$

and

$$n = 1 \cdot 40 \, . \, u \sqrt{\frac{p \, . \, a \, . \, v}{\text{S.H.P.}(1 - s)}} \text{ metric units,}$$

$$n = \cdot 936 \, . \, u \sqrt{\frac{p \, . \, a \, . \, v}{\text{S.H.P.}(1 - s)}} \text{ English units.}$$

If we take as an example the same values as in § 112 we have:

$v = 33$ knots,
S.H.P. $= 4{,}500$,
$u = 70$ m., or 230 ft.,
$a = \cdot 56$,
$s = 20$ per cent.,
$1 - s = \cdot 8$,
$p = 12{,}900$ kg. per sq. m., or 2,641 lbs. per sq. ft.

$$n = 1 \cdot 40 \, . \, 70 \sqrt{\frac{12{,}900 \, . \, \cdot 56 \, . \, 33}{4{,}500 \, . \, \cdot 8}} = 798, \text{ say } 800, \text{ metric units,}$$

$$n = \cdot 936 \, . \, 230 \sqrt{\frac{2{,}641 \, . \, \cdot 56 \, . \, 33}{4{,}500 \, . \, \cdot 8}} = 791, \text{ say } 800, \text{ English units.}$$

§ 114. **Constructive Development of the Propeller.**—The calculation of the strength of the propeller blade is the same as for reciprocating engines, except that special attention must be given to centrifugal force. For that reason the propeller is designed with the centre line of the blades at right angles to the axis.

The material used is always bronze, either Mangan bronze or lately Rübel bronze, which has proved to be of special strength, and highly suitable for use in salt water.

The propellers are usually so small that the blades boss can be cast in one, and only rarely are the propellers of such size that the blades have to be bolted to the boss.

The shape of the boss must be such that the water flowing to the propeller is disturbed as little as possible, and the boss, therefore, is generally made oblong instead of spherical.

PART VI.

CONDENSING PLANT.

§ 115. **General Remarks.**—With the surface condensing plant used with reciprocating engines, a vacuum of 85 to 90 per cent. was usually considered to be sufficient, and no difficulty was experienced in obtaining it. A higher vacuum was not aimed at, because the increase in output of the plant due to the increase in vacuum was not sufficiently great to warrant the increased capital expenditure and pumping power required to enable the high vacuum to be obtained, and because of the low temperature of the condensed steam water resulting with high vacua. A material reduction in the temperature of the feed water would result in a higher coal consumption so that the gain in vacuum would have been doubtful.

For a steam turbine plant on the other hand a high vacuum is essential, as the increase in output due to a low back pressure is considerably greater in a steam turbine than in a reciprocating engine (§ 21).

It has therefore been the aim of designers of turbine plant to perfect the condensing apparatus as much as possible, both in the interests of a high vacuum and to avoid at the same time the cooling of the condensed steam water below the temperature corresponding to the vacuum.

To obtain a good vacuum, the following conditions are necessary:—

1. Ample supply of cooling water.
2. Rapidity in condensing of the steam.
3. Rapid and perfect removal by the air pump of the air entering the condenser.

§ 116. **Quantity of Cooling Water.**—The outlet temperature of the cooling water is usually only 3° to 5° C. (5·4° to 9° F.) below the temperature corresponding to the vacuum in the condenser

The following symbols will be used in the calculations given below:—

Q_h. Steam condensed per hour, kg. metric, lbs. English.

S. Cooling surface, sq. m. metric, sq. ft. English.

W. Quantity of cooling water per hour, kg. metric, galls. English.

K. Heat transmission coefficient.

p_c. Pressure at condenser inlet, kg. per cm. sq. metric, lbs. per sq. in. English.

t_i. Cooling water inlet temperature, ° C., ° F.

t_o. Cooling water outlet temperature, ° C., ° F.

t_c. Temperature corresponding to p_c, ° C, ° F.

If we denote the difference between the temperature corresponding to the pressure in the condensers, and the outlet temperature of the cooling water $(t_c - t_o)$ as the drop, and assume this to be on an average 4° C., or say 7° F., then $t_o = t_c - 4°$ C., or $t_o = t_c - 7°$ F.

The increase in the temperature of the cooling water is:—

$$t_o - t_i = t_c - t_i - 4° \text{ C., or} = t_c - t_i - 7° \text{ F.}$$

If a vacuum of 96 per cent. is assumed, corresponding to $p_c = {\cdot}04$ kg. per sq. cm., or ·589 lbs. per sq. in., the corresponding temperature will be $t_c = 29°$ C., or $t_c = 84{\cdot}5°$ F.

One kilogram steam at this temperature has a total heat of about 608 cal. (or 1 lb. of steam 1,097 B.Th.U.), so that 608 - 29 = 579 cal. (or 1,097 - 53 = 1,044 B.Th.U.) have to be withdrawn.

The necessary quantity of cooling water per kg. (or lbs.) of steam will be:—

$$W = \frac{579}{t_o - t_i} = \frac{579}{t_c - t_i - 4} \text{ kg.,}$$

or,

$$W = \frac{1{,}044}{t_o - t} = \frac{1{,}044}{t_c - t_i - 7} \text{ lbs.}$$

As the annual mean temperature of sea water in our regions is about 15° C. (59° F.), the quantity of cooling water required to condense a unit weight of steam at the above vacuum will be:—

$$W = \frac{579}{29 - 15 - 4} = 32{\cdot}1 \text{ kg.,}$$

or,

$$W = \frac{1{,}044}{84{\cdot}5 - 59 - 7} = 32{\cdot}1 \text{ lbs.}$$

§ 117. **Cooling Surface.**—The following requirements must be fulfilled to obtain rapid condensation :—

1. A minimum temperature of the cooling surface.
2. A certain amount of cooling surface.
3. Efficient distribution of the steam over the cooling surface, and
4. Rapid drainage of the condensed steam water from the surface of the tubes to the lowest point of the condenser.

To maintain a low temperature of the cooling surface it is advisable to pass the water through the tubes at a high velocity.* The reason for this is that the water is kept in a constant eddying motion, and prevents in the tube the formation of cylindrical streams of water, the periphery of which streams alone come in contact with the tubes. Experiments have shown that the velocity of the water through the tubes should be about 1·8 to 2·5 m. per second, or 6 to 8 ft. per second.

The total friction head on the circulating water pump will then be about 5 to 8 m., or say 16 to 26 ft., which will have to be taken into consideration when designing the pump.

From data obtained with condensers actually in service the amount of cooling surface can be fixed so as to condense 36 to 60 kg. per sq. m. (or about 7·3 to 12 lbs. per sq. ft.).

In the following a method is given which will enable the cooling surface to be calculated and which will give approximately correct values. Referring to the above symbols the following relation holds good :—

$$Q_h \,.\, L = S \,.\, K\left(t_c - \frac{t_i - t_o}{2}\right).$$

Numerous experiments, amongst which are those of Professor Josse (see footnote below), have shown that the heat transmission coefficient K depends more upon the velocity of the water through the tubes than upon the material of which the tubes are made.

In marine installations the heat transmission coefficient varies between 2,200 and 3,200 metric units, or 450 to 654 English units, at a velocity of the cooling water through the tubes of 1·8 to 2·5 m. per sec., or 6 to 8 ft. per sec., the higher value corresponding to the higher water velocity. These values of the heat transmission coefficient of course assume that the condenser is correctly constructed, and especially that ample means are provided for withdrawing the air. If this is not the case lower values of the coefficient will have to be taken.

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* Professor Weighton, "The Efficiency of Surface Condensers," Institute of Naval Architects, 1906. Professor Josse, "Jahrbuch der Schifferbau Technischen Gesellchaft," 1909.

In the following a concrete example is given.

For a turbine installation on a torpedo boat it is assumed that:—

The quantity of exhaust steam -	$Q_h = 40{,}000$ kg., or 88,300 lbs.
The vacuum - - - - -	90 per cent.
The absolute condenser pressure -	$p_c = \cdot 1$ atm., or 1·47 lbs. per sq. in.
The temperature corresponding to p_c	$t_c = 46°$ C., or 115° F.
Inlet temperature of cooling water -	$t_i = 16°$ C., or 61° F.
Outlet " " " -	$t_o = 27°$ C., or 80·5° F.
The heat transmission coefficient -	$K = 2{,}600$, or 532 English units,

corresponding to a water velocity through the tubes of 2·0 m. per sec., or 6·6 ft. per sec.

The cooling surface for the above formula will be—

$$S = \frac{Q_h \,.\, L}{\left(t_c - \frac{t_i + t_o}{2}\right)K}.$$

$$S = \frac{40{,}000 \,.\, 568}{\left(46 - \frac{16+27}{2}\right)2{,}600} = 356 \text{ sq. mm.,}$$

or,

$$S = \frac{88{,}300 \,.\, 1{,}027}{\left(115 - \frac{61+80{\cdot}5}{2}\right).\, 532} = 3{,}860 \text{ sq. ft.}$$

The distribution of the steam at the inlet to the condenser, which in turbine condensing plant is already a good one on account of the large area of the exhaust branch, is usually increased by widening the inlet to the condenser so that in some cases this occupies nearly the whole length of the condenser. To reduce the resistance against the incoming steam a number of condenser tubes opposite the inlet are removed (Fig. 79).

The drainage of the condensed steam from the tubes is difficult to accomplish as the water condensed in the higher tubes drops on to the lower ones, surrounding the latter with a film of water of comparative thickness, which seriously affects the heat transmission rate. In the usual surface condensers the condensation of the steam does, therefore, not occur on the surface of the tubes, but on the cooled film of water surrounding the tubes.

It is possible that condensers with vertical or inclined tubes would allow of a more rapid drainage of the tube surface, but such an arrangement would probably be difficult to place on board a vessel.

In small condensers, and when space permits in larger installations, the condenser is made so long as to allow of a single passage of the

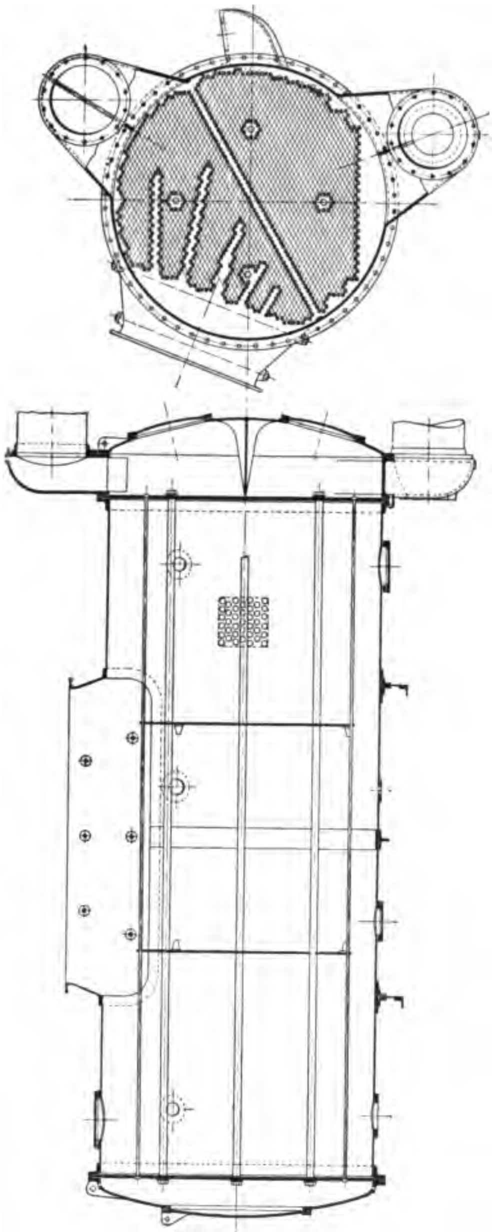


Fig. 79.

cooling water. In shorter condensers it is necessary, in order to obtain the requisite water velocity, to pass the water two or three times through the condenser. The water heads and doors are then provided with suitable baffle plates, and so designed that the coldest water enters the condenser at the bottom leaving the condenser at the top.

§ 118. **Withdrawal of Air.**—In order that the cooling surface should have a maximum efficiency, and to obtain a high vacuum with a maximum efficiency of the cooling surface, the air contained in the condenser must be removed as completely and as rapidly as possible.

Let p_c be the absolute pressure in the condenser as measured by the vacuum gauge,

p_v the absolute vapour pressure in the condenser corresponding to the temperature of the condensed steam water in the condenser,

p_a the absolute pressure of the air in the condenser.

Then in accordance with the law of Dalton :—

$$p_c = p_v + p_a.$$

If, further—

w_a is the quantity of air at atmospheric pressure entering the condenser per hour at a temperature equal to that of the condensed steam water in cub. m. (cub. ft.), and

v is the volume dealt with by the air pump per hour in cub. m. (cub. ft.),

the air pump would create a pressure when dealing with air only of

$$p_a = \frac{w_a}{v}.$$

As the air contains vapour, the pressure created in the condenser will be—

$$p_c = p_v + \frac{w_a}{v},$$

and the necessary hourly capacity of the air pump—

$$v = \frac{w_a}{p_c - p_a}.$$

In the case of a wet-air pump this capacity will have to be increased by the volume of the feed water.

The formula indicates that for a given pressure in the condenser p_c, the volume of the air pump v will be a minimum for a minimum vapour pressure, or for a minimum temperature of the air and vapour mixture. With the use of wet-air pumps, the condensed steam water for this reason is cooled to a temperature as far as possible below the temperature corresponding to the pressure in the condenser. This cooling, however, presents in every case a thermal loss.

The use of dry-air pumps dealing with the mixture of air and vapour only, avoids this loss. The condensed steam water can be removed by a separate pump at a temperature corresponding to the pressure in the

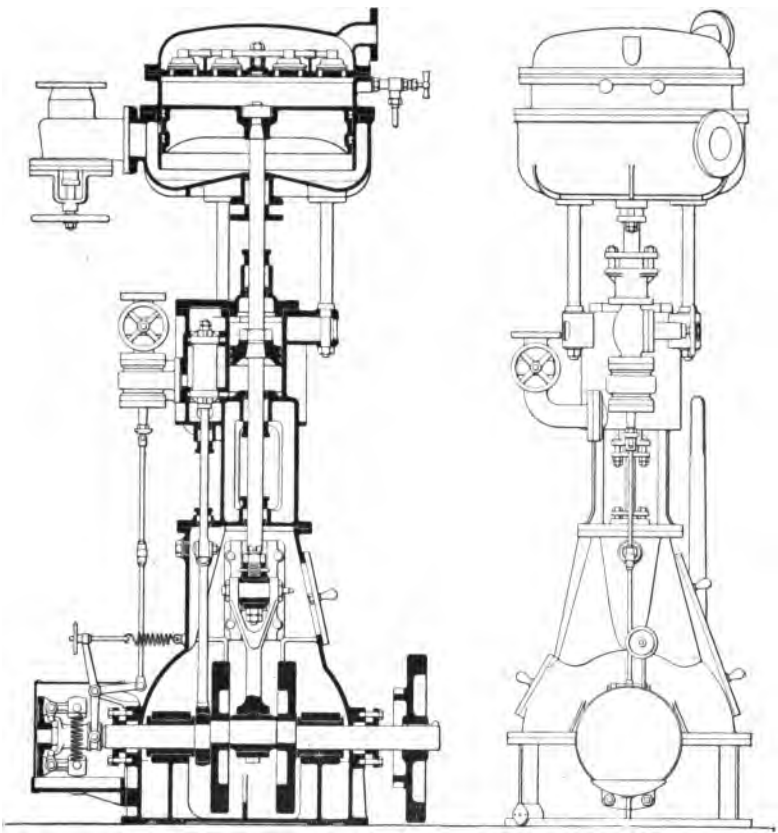


Fig. 80.

condenser, and the air and vapour is withdrawn at the coldest point in the condenser. For this purpose part of the tubes near the cooling water inlet are used as an air cooler, to which the suction pipe of the dry-air pump is connected.

The quantity air w_a entering the condenser depends upon the air tightness both of the glands of the turbines and of the whole system;

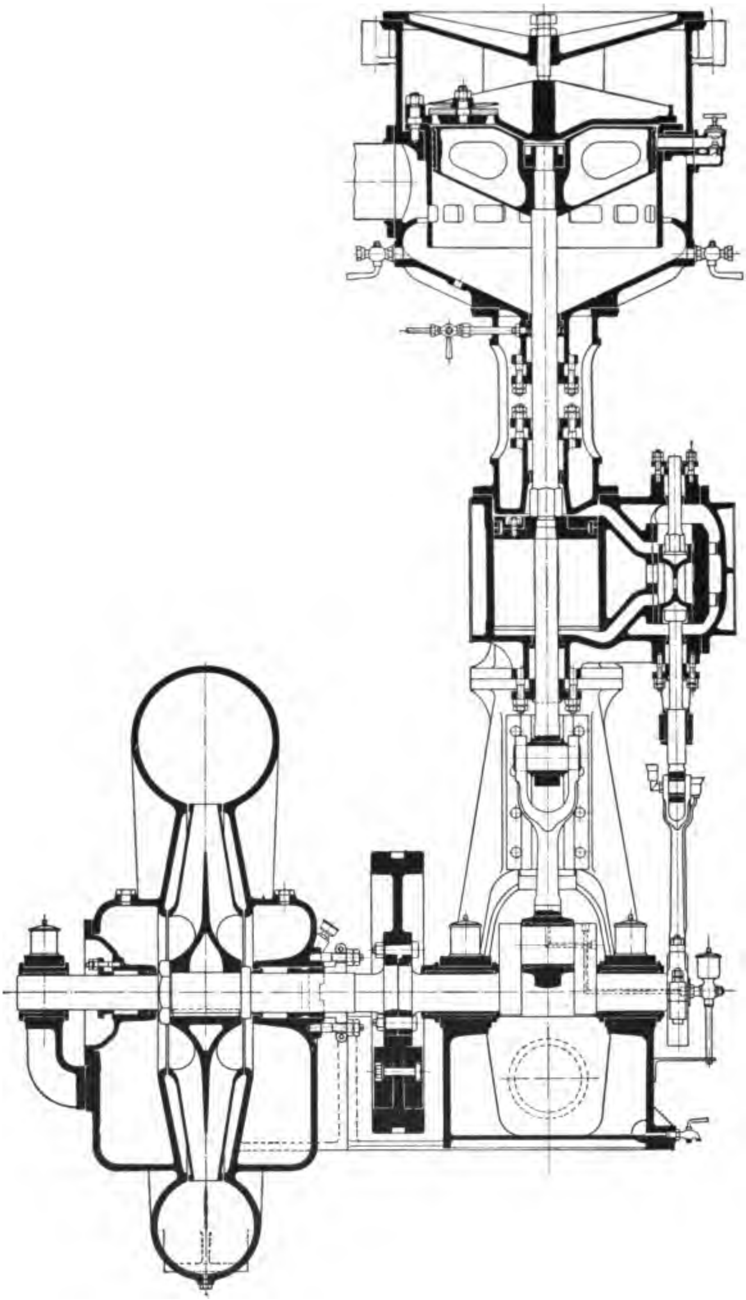


Fig. 81.

measurements made in actual installations have given results of ·0015 to ·005 cub. m. (·053 cub. ft. to ·176 cub. ft.) per hour per B.H.P.

§ 119. **Air Pump**—*Wet-Air Pumps.*—It is usual to employ single-acting double-cylinder wet-air pumps similar to those in use for large reciprocating engines for removing the condensed steam water as well as the air and vapour from the condenser. The pumps should be mounted about 2 ft. below the lowest point of the L.P. turbine to allow the turbine to drain into the air pump (see § 95).

Wet and Dry-Air Pumps.—By dealing separately with the air and condensed steam water a large volume of air can be dealt with, and in such cases a dry-air pump is required for dealing with the air and vapour only. The condensed steam is removed from the lowest point of the condenser by a comparatively small duplex steam pump working at twenty-five to forty strokes per minute, delivering the water to a feed heater or the hotwell tank. It is, however, preferable to instal a wet-air pump to take the place of the duplex pump, as the former serves as a standby in case of a breakdown to the dry-air pump.

Fig. 80 shows a steam-driven dry-air pump, and Fig. 81 a similar pump combined with the circulating water pump. The air pumps are generally of the Edwards type, and fitted with delivery valves only. The pumps are cooled and water-sealed by admitting a small amount of water to either side of the plunger (Fig. 81), and draining the delivery space. In some cases the cooling water is only admitted into the compression chamber, causing indirect cooling of the air. If the dry-air pump is designed as a separate pump (as the Weir pump shown in Fig. 80), it can be placed in any convenient place in the engine-room or neighbourhood, but care must be taken in the lay-out of the pipe line connecting the pump to the condenser that the pipe line is of ample size, short, free of air locks and sharp bends. The engine driving the pump must be fitted with a governor to prevent the pump racing at high vacua.

The normal number of revolutions of these dry-air pumps is 200 to 300 revs. per min.

PART VII.

ARRANGEMENT OF THE TURBINES.

SECTION I.

Parsons Turbines in Torpedo Boats.

§ 120. **General Remarks.**—The general arrangement of the turbines in a torpedo boat is to have three shafts with a high-pressure turbine driving the centre shaft, and a low-pressure turbine on each wing shaft. Although this is the general disposition of these turbines, there are certain variations possible in the arrangement.

§ 121. **Torpedo Boats with Parsons Turbine with Engine-Room divided by a Bulkhead.**—The arrangement shown in Fig. 82 is designed from the point of view of being able to work one-half of the plant should the other be inaccessible. The high-pressure turbine drives the centre shaft, and receives boiler steam through the valve operated by the hand-wheel *k*. Each low-pressure turbine contains a reversing turbine worked by high-pressure steam. The low-pressure turbines can also be worked with high-pressure steam for ahead motion, so that each outer shaft can be worked ahead or astern by suitable two-way valves. In the arrangement shown, the two-way valve of the after turbine on the starboard side can be worked from the starting platform in the forward engine-room. The hand-wheels *n* and *o* operate these two valves, the valve *m* serving to close the steam mains to both the two-way valves. Non-return valves are fitted in the pipe connections between the high-pressure and the two low-pressure turbines. This is to prevent boiler steam entering the exhaust end of the high-pressure turbine when the low-pressure turbines are running with boiler steam. On the port-side shaft a cruising turbine is fitted. When cruising, the steam is first admitted to the cruising turbine, from there to the high-

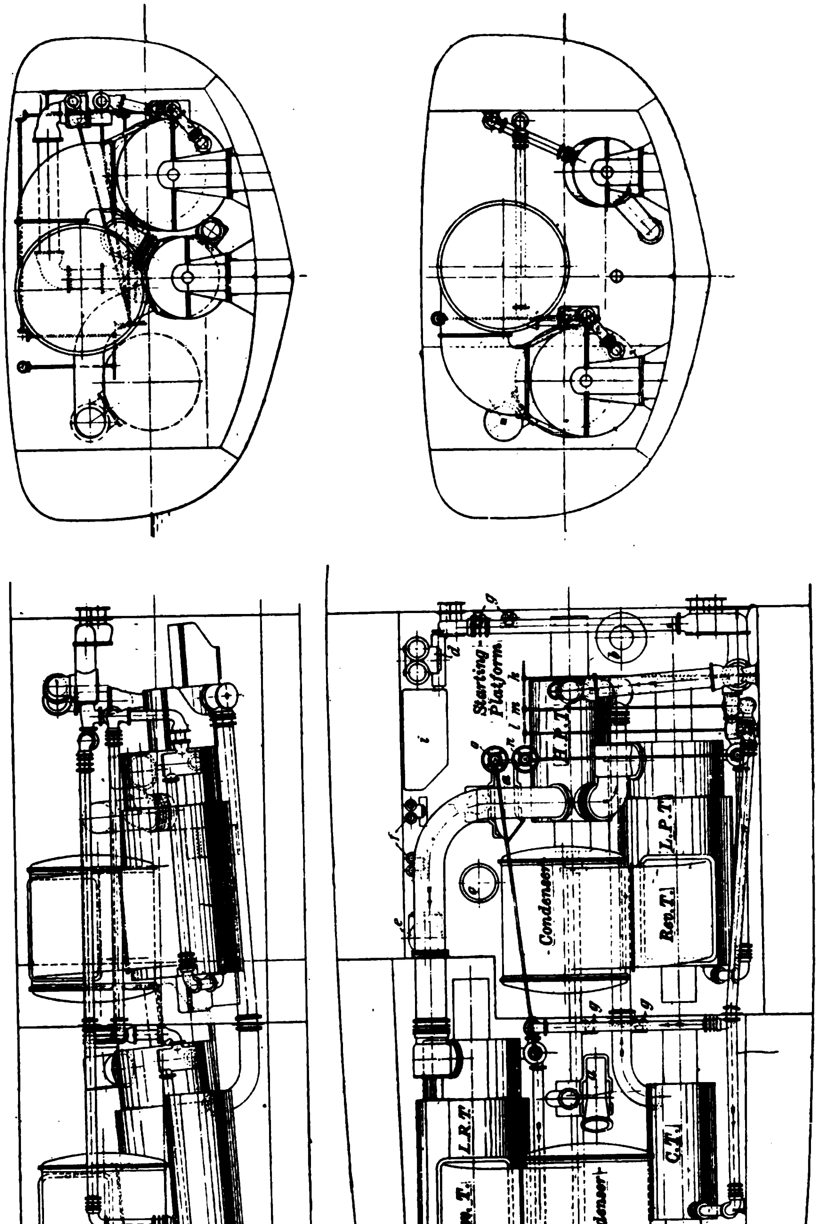

Starting Platform
H.P.T.
L.P.T.
Condenser
Rev. T.
L.R.T.
Rev. T.
Condenser
C.T.

Fig. 82.

pressure turbine and then to the low-pressure turbines. The steam admission to the cruising turbine is through the valve operated by the hand-wheel *l*.

Each engine-room contains:—

One circulating water pump - - -	*a*
One wet-air pump - - - -	*b*
One dry-air pump - - - -	*c*
One main feed pump - - - -	*d*
One evaporator - - - - -	*e*
Two oil pumps (one spare) - - -	*g*
One lighting set - - - -	*h*
One hot well - - - - -	*i*

and two bilge pumps *f* are placed in the forward engine-room.

§ 122. **Torpedo Boat without Bulkhead in Engine-Room** (Fig. 83).—This arrangement of the engine-room is very much simpler and the machinery more accessible than the one described above, but has the disadvantage of being entirely inaccessible if flooded or filled with steam.

The steam admission to the high-pressure turbine is controlled by the hand-wheel *m*. The turbines on the two outer shafts are controlled by *l*. These hand-wheels operate two-way valves, allowing boiler steam to be admitted to the two low-pressure, or to the two reversing turbines in the low-pressure turbine casing.

To maintain high economy at cruising speeds, a cruising turbine, I.P.C.T., is coupled to the starboard shaft. If the economy is to be maintained at still lower speeds, a high-pressure cruising turbine, H.P.C.T., is coupled to the shaft on the port side. The high-pressure and the intermediate cruising turbines work in series, the steam admission being controlled by the hand-wheels *i* and *k*.

After passing through both cruising turbines, the steam flows through the high-pressure main turbine, and finally through the two low-pressure turbines. Non-return valves are fitted between the high-pressure and the intermediate-pressure cruising turbine, and between the high-pressure main turbine and the low-pressure turbine, the former to prevent steam entering the cruising turbine when boiler steam is admitted to the high-pressure main turbine, and the latter to prevent steam entering the high-pressure turbine when the low-pressure turbine is working with boiler steam.

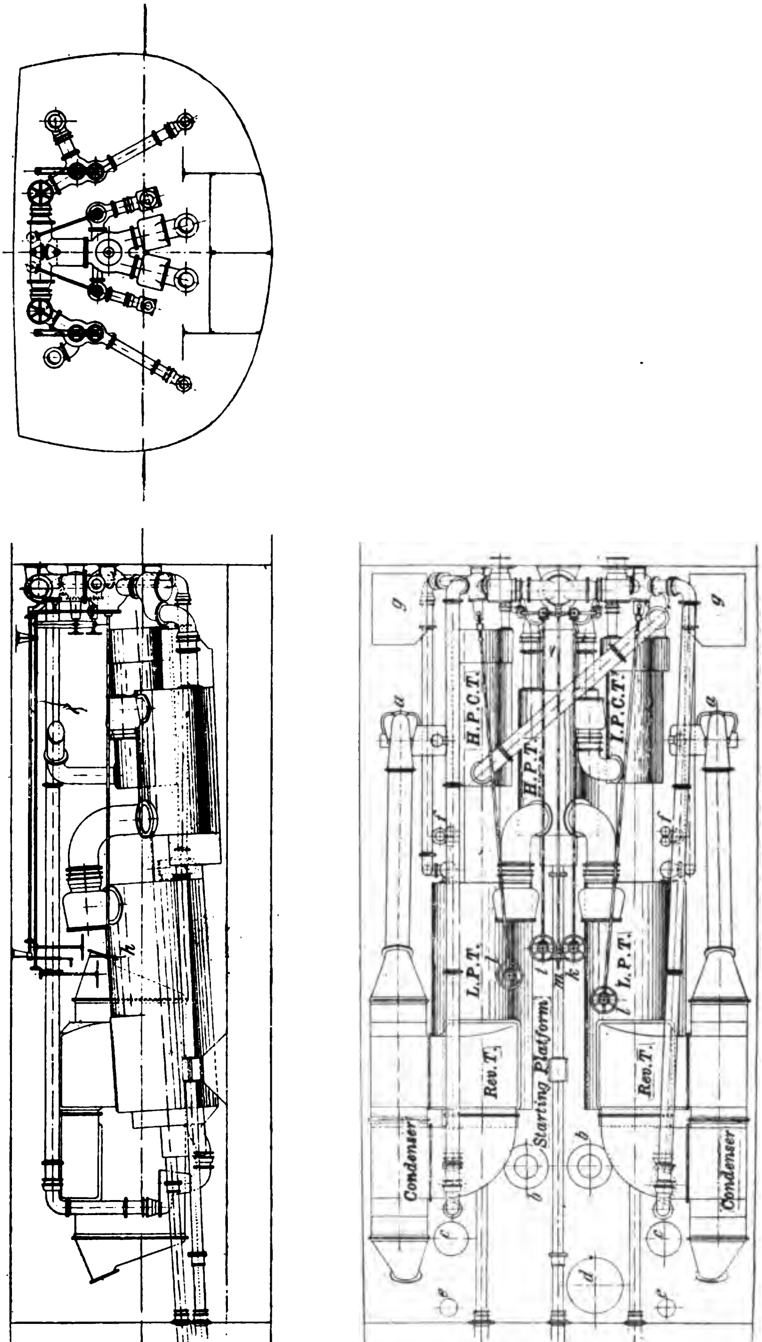

H.P.C.T.
I.P.C.T.
H.P.T.
L.P.T.
L.P.T.
Rev.T.
Rev.T.
Starting Platform
Condenser
Condenser

Fig. 83.

In addition to the turbines, the engine-room contains the following auxiliaries :—

Two circulating water pumps - - -	*a*
Two wet-air pumps - - - -	*b*
Two dry-air pumps - - - -	*c*
One evaporator - - - - -	*d*
Two bilge pumps - - - -	*e*
Two oil pumps - - - - -	*f*
Two hot-well tanks - - - -	*g*

The space available for the condenser enables a condenser with a single pass to be installed (see § 117). There is sufficient space between the condenser and circulating water pump for withdrawing tubes.

SECTION II.

Torpedo Boat with Single-Shaft Turbine of the Curtis and A.E.G. Type.

§ 123. **General Remarks.**—The arrangement of the Parsons turbines is somewhat complicated on account of the connections between the individual turbines, of which two are arranged to work in series on the same shaft. As the single-shaft turbines of the A.E.G. and Curtis type are very simple in arrangement, they are particularly suitable for torpedo boats.

§ 124. **Torpedo Boat with A.E.G. Turbines with Engine-Room divided by a Bulkhead** (Fig. 84).

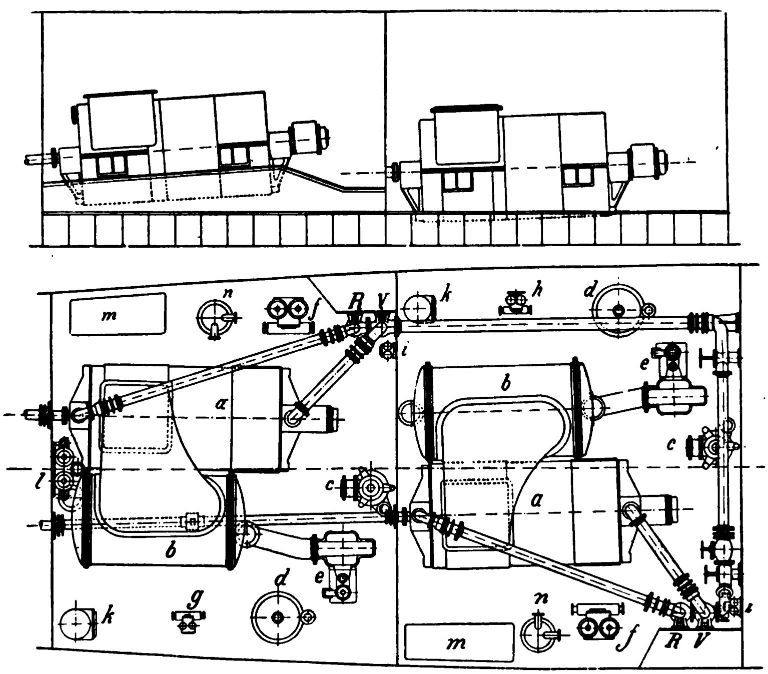


Fig. 84.

Each half of the engine-room contains a single turbine, combining the main and reversing turbine in the same casing *a*. In addition each half contains a condenser *b*, a complete set of air pumps, circulating water pumps and auxiliaries, making each half of the engine-room a complete and independent unit.

The valve V is the main ahead, and valve R the reversing valve. To reverse, it is only necessary to close the one and open the other valve (see § 94). For economy in cruising a bye-pass valve or a cruising valve is provided. These valves do not influence the simplicity of manœuvring in the least, as they do not increase the number of operations when reversing, nor do they influence the interdependence of the turbine and shaft.

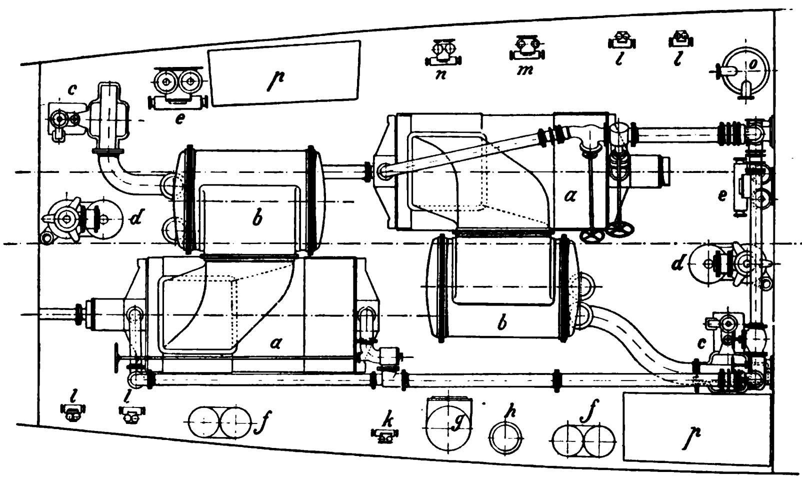


Fig. 85.

Each half of the engine-room contains :—

One circulating water pump - - -	*b*
One wet-air pump - - - -	*c*
One dry-air pump - - - -	*d*
One main feed pump - - - -	*f*
One oil pump - - - - -	*i*
One hot-well tank - - - -	*m*
One evaporator - - - - -	*k*
One cooling water pump - - -	*g*
One bilge pump - - - -	*h*

The duplicate feed pumps are placed in the stoke-hold. The circulating water passes through the condenser once.

The starting platform is so arranged that the main valves and auxiliaries, especially the oil pump, are easily accessible. The gauge-glass of the hot-well tank should be visible from the starting platform.

§ 125. **Torpedo Boat with A.E.G. Turbines without Bulkhead.**—This arrangement occupies even less space than that described above, but has the disadvantage of inaccessibility of the whole machinery in case of accident.

Referring to Fig. 85 the engine-room contains the following machinery :—

Two complete turbine sets, each consisting of a high-pressure, low-pressure, and reversing turbine in one casing - - -	*a*
Two condensers - - - -	*b*
Two circulating pumps - - - -	*c*
Two wet-air pumps - - - -	*d*
Two main feed pumps - - - -	*e*
Two dry-air pumps - - - -	*f*
One evaporator - - - -	*g*
One soft-water filter - - - -	*h*
One evaporator pump - - - -	*k*
Four oil pumps (two spare) - - -	*l*
One bilge pump - - - -	*m*
One cooling water pump - - -	*n*
One feed heater - - - -	*o*
Two hot-well tanks - - - -	*p*

SECTION III.

Cruisers and Battleships with Parsons Turbines.

§ 126. The four-shaft arrangement is used in both types of vessels (Figs. 86 and 87). The whole plant is divided into two symmetrical parts, each with two shafts. The two outer shafts are each driven by high-pressure main turbines, and the inner shafts by the low-pressure turbines. If cruising turbines are installed, these drive the inner shafts, either a high-pressure cruising turbine on each shaft, or a high-pressure cruising turbine on one shaft and a low-pressure cruising turbine on the other. The reversing turbines are arranged either one on each wing shaft, so that these shafts alone are reversible, or a reversing turbine is in addition fitted inside each low-pressure main turbine. In the latest arrangement the reversing turbines in the low-pressure main turbines are built as low-pressure turbines, and work in series with the high-pressure reversing turbines.

Fig. 86 shows such an arrangement for a small cruiser where each half is divided by a fore-and-aft bulkhead.

At full speed, which in this case is 23 knots, the steam first flows through the high-pressure main turbine driving the outer shaft, from here to the low-pressure main turbine on the inner shaft and to the condenser. At reduced speed, from 20 knots downwards boiler steam is admitted to the low-pressure cruising turbine on the inner port shaft. From here the steam passes to the two high-pressure main turbines, then to the two low-pressure main turbines and to the condenser. If the speed is reduced below 16 knots the steam before entering the low-pressure cruising turbine is admitted to the high-pressure cruising turbine on the inner starboard shaft.

Non-return valves are fitted in the steam-pipe connections between the high-pressure and the low-pressure cruising turbines, as well as between the low-pressure cruising and the high-pressure main turbines. The valves are necessary to enable the turbines which are not under steam—that is in one case the high-pressure cruising turbine, and in the other case the low-pressure cruising turbine—to rotate in a vacuum, and thus to minimise the resistance. A disadvantage of this arrangement is the complicated piping and the lack of simplicity. A further disadvantage is that at full speed two shafts, and at cruising speeds all four shafts, are dependent on one another.

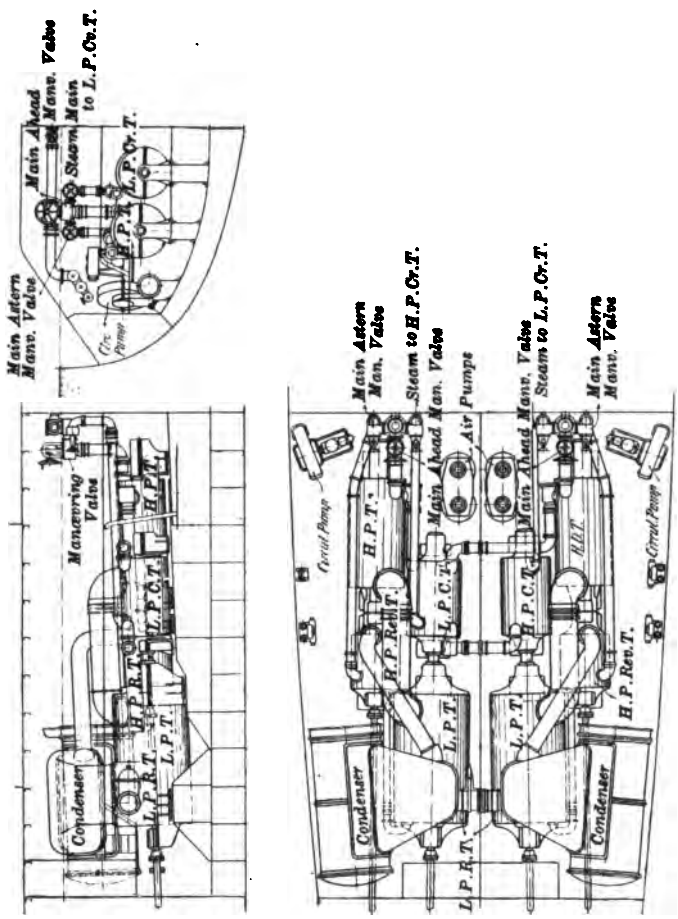
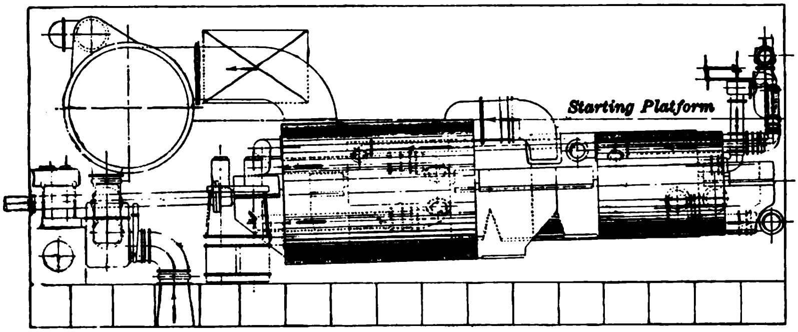


Fig. 86.

Fig. 86 further shows the arrangement of the auxiliaries. Each half of the engine-room contains:—

A circulating water pump.
A two-throw air pump.
An oil pump.
A cooling water pump.
A bilge pump.

The other auxiliaries such as the lighting set, evaporator, feed pumps, are mounted in a separate space.



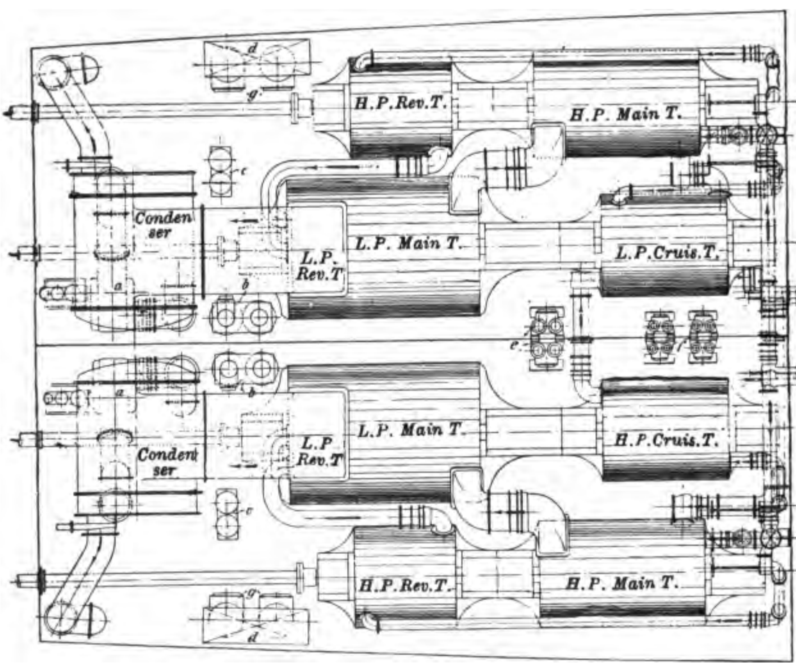


Fig. 87.

The arrangement for battleships with Parsons turbines, shown in Fig. 87, is similar to that of a cruiser described above.

SECTION IV.

SOME NOTES ON THE USE OF CRUISING TURBINES IN WAR VESSELS FITTED WITH PARSONS TURBINES.

§ 127. Lately the use of separate cruising turbines in vessels fitted with Parsons turbines has not been so frequent, and this appears to be the first step to approach the single-shaft arrangement.* The following are probably the reasons why these turbines are being omitted :—

1. The most frequent accidents with Parsons turbines have been blade strips in consequence of the small blade clearances necessary in the pure reaction type of turbine. As the amount of steam admitted to the cruising turbine is small, the blades are short, and to reduce the leakage losses as much as possible the clearances are naturally made small, as otherwise the losses are considerable.

2. The cruising turbines when the vessel is steaming ahead rotate in a vacuum requiring an increased air pump capacity. The increased number of glands also increase the possibilities of air leakage.

3. The efficiency of the cruising turbine at low speed is frequently reduced as the steam, after passing through the cruising and high-pressure turbines, has expanded to such a degree that the low-pressure turbine is only running idle.

4. The cruising turbine takes up a lot of space which could be utilised more advantageously for other purposes.

5. The cruising turbines cause considerable complication of the pipework.

6. The turbines and pipework together represent a large dead weight, which if used for coal storage, would compensate for part of the economy attained with these turbines.

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* See Stodola, 4th German Edition, pp. 519 ff.

SECTION V.

CRUISERS AND BATTLESHIPS WITH A.E.G. AND CURTIS TURBINES.

§ 128. **Cruisers with A.E.G. Turbines.**—Both in cruisers and battleships the arrangement, either with A.E.G. or Curtis turbines is similar, in so far that each shaft is driven independently of the other by a separate set of turbines. This allows a subdivision of the engine-room into two, or, as in the latest battleships, into three separate spaces. Fig. 88 shows the arrangement of a small cruiser. The engine-room is divided by a fore-and-aft bulkhead into two equal spaces, each containing one high-pressure and one low-pressure turbine. The cruising turbine is built into the casing of the low-pressure turbine. Each half of the engine-room contains:—

One condenser.
One circulating water pump - - - *a*
One wet-air pump - - - - *b*
One dry-air pump - - - - *c*
Two oil pumps.
One pump for circulating the oil cooler water, and
One bilge pump.

The starting platform has two valves, one for ahead motion with hand-wheel *d*, and one for astern motion with hand-wheel *e*. The simplicity of this arrangement compared with that of the Parsons arrangement (Fig. 86) is at once apparent.

§ 129. **Cruiser with Curtis Turbine.**—Fig. 89 is a plan view

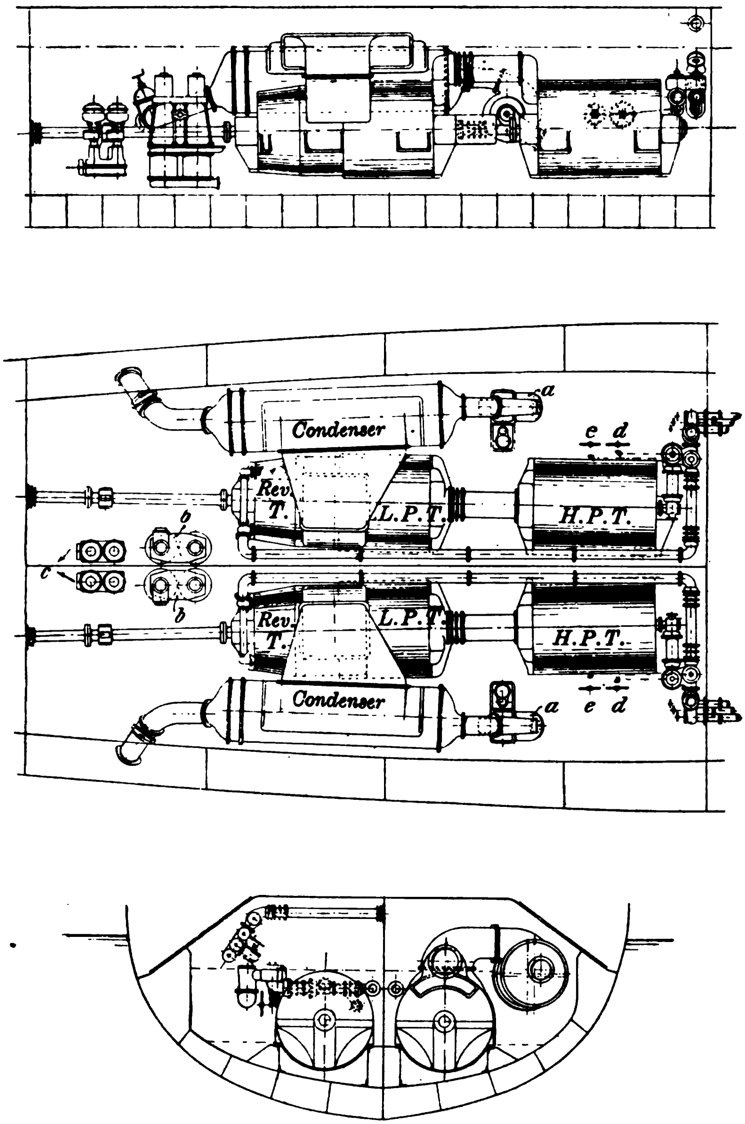


Fig. 88.

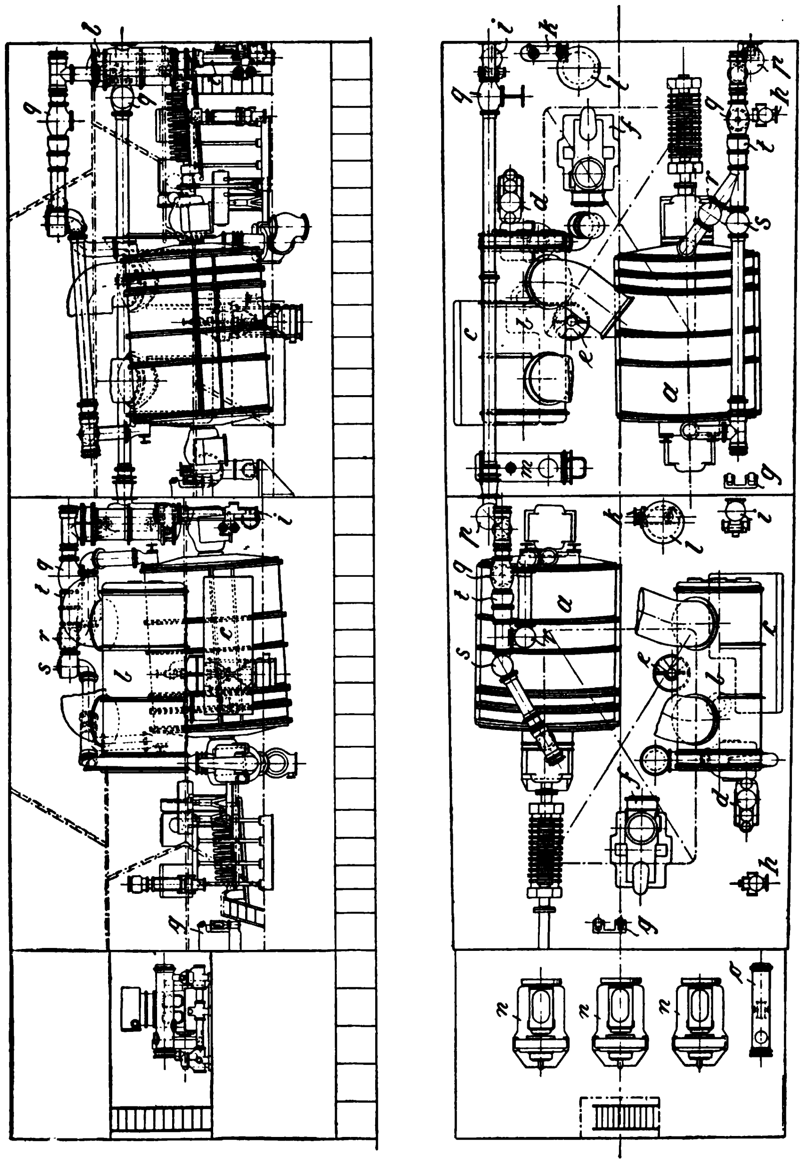


Fig. 89.

and elevation of the turbine equipment of a cruiser. Each of the two spaces contains :—

One main and reversing turbine	*a*
One condenser	*b*
One hot-well tank	*c*
One circulating water pump	*d*
One wet-air pump	*e*
One dry-air pump	*f*
Two barring engines	*g*
One bilge pump	*h*
One feed pump	*i*
Two oil filters	*k*
One feed heater	*l*

The equipment further consists of :—

One auxiliary condenser	*m*
Three generating sets	*n*
One condenser for the generating sets	*o*

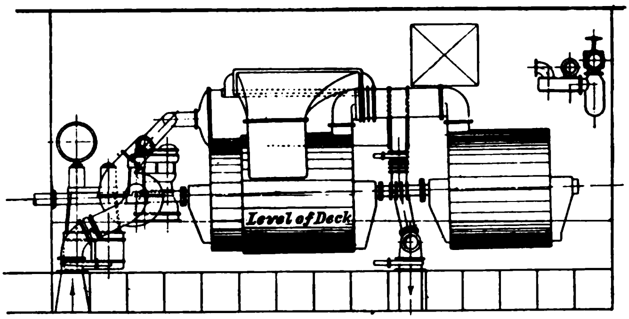
The sets are worked as follows :—The steam from the boilers passes through a separator *p*, a main steam valve *q*, a butterfly valve *t*, passing either through the starting valve *s* to the ahead turbine or the valve *r* to the reversing turbine.

A characteristic of this arrangement is not only the symmetry of each half, but that the two halves are exactly the same, the reversing turbine being arranged aft, and the thrust block forward in the forward space, the reverse being the case in the after space.

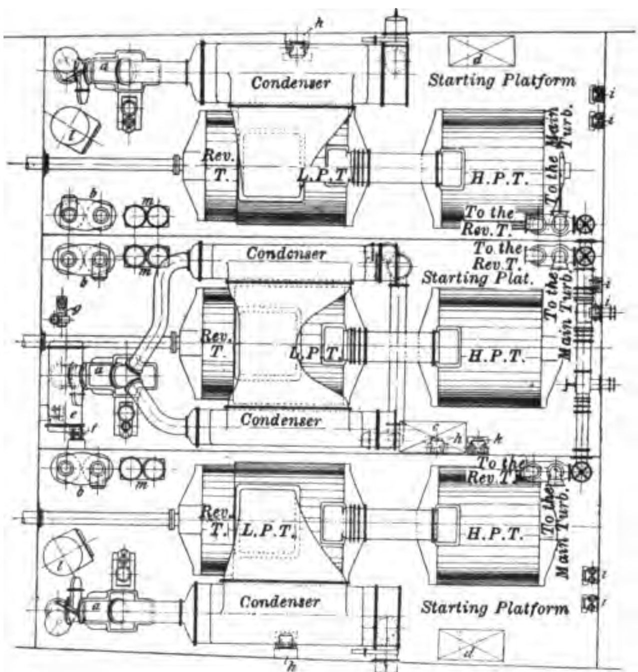
§ 130. **Battleship with A.E.G. Turbine.**—The equipment of a battleship with A.E.G. Curtis turbines is shown in Fig. 90. The only difference between this and the equipment of the cruiser (§ 128) is that the engine-room of the battleship is divided into three spaces, each with one set of turbines in place of two spaces with two turbines.

Each hold contains the following :—

One circulating water pump	*a*
One wet-air pump	*b*
One dry-air pump	*m*
Two oil pumps	*i*
One hot-well tank	*d*, *d*, and *c*



Long. Section



Plan

Fig. 90.

The auxiliary condenser *e* with circulating water pump *g*, air pump *f*, and a small feed pump is mounted in the centre section, and consequently the hot-well tank *c* is not fitted with a feed water purifier. A bilge pump *k* and evaporator *l* are fitted.

Each turbine set is fitted with a separate ahead and astern valve, as shown in the illustration.

SECTION VI.

§ 131. **Passenger Steamers with Parsons Turbines, Triple Shaft Arrangements** (Figs. 91 and 92).—The high-pressure turbine

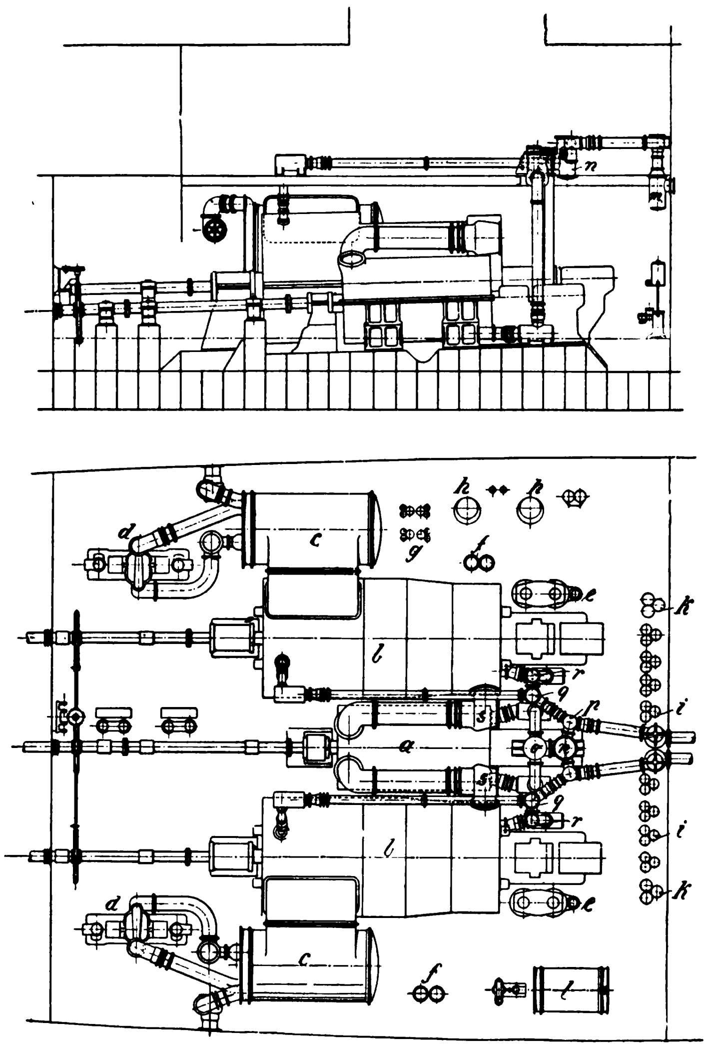


Fig. 91.

drives the centre shaft *a*. Each wing shaft is driven by a low-pressure and a reversing turbine, both contained in the same casing *b*. At full speed the steam is taken in two separate pipe lines from the boilers, and after passing the separators *m* and the valve *p*, to the strainer *n* and butterfly valve *o*. From here the steam passes through two pipe lines to the high-pressure turbine *a*. From the exhaust of the high-pressure turbine the steam, after passing the valves *s* on the low-pressure turbine casing, is led to the low-pressure turbine and the condenser *c*.

When entering or leaving a harbour, or cruising in home waters where much manœuvring is necessary, the valves *s* are closed, and

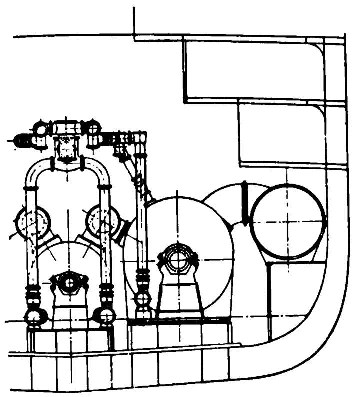


Fig. 92.

the high-pressure turbine put out of commission. Steam is then admitted through the valves *p*, and the two-way valves *q*, either through the pipe line *v* to the low-pressure turbine, or to the reversing turbine in the low-pressure turbine casing. The circulating water pump *d* is placed behind the condensers, the two wet-air pumps *e*, and the dry-air pump *f* close to the low-pressure turbine. On the port side two evaporators *h* and an oil pump *g* are fitted. The auxiliary condensing plant *l* is at the starboard. Close up to the bulkhead are the main feed pumps *k*, also the bilge cooling, fire, and fresh water pumps *i*. There is one barring gear for all three shafts, driving a shaft athwartships and provided with the necessary couplings for connection to any of the three worm-wheels on the main shafts.

§ 132. **High Speed Steamer with Parsons Turbines.**—The most modern examples of such a vessel are the two Cunarders, and the following is a short description of the turbine installation of the

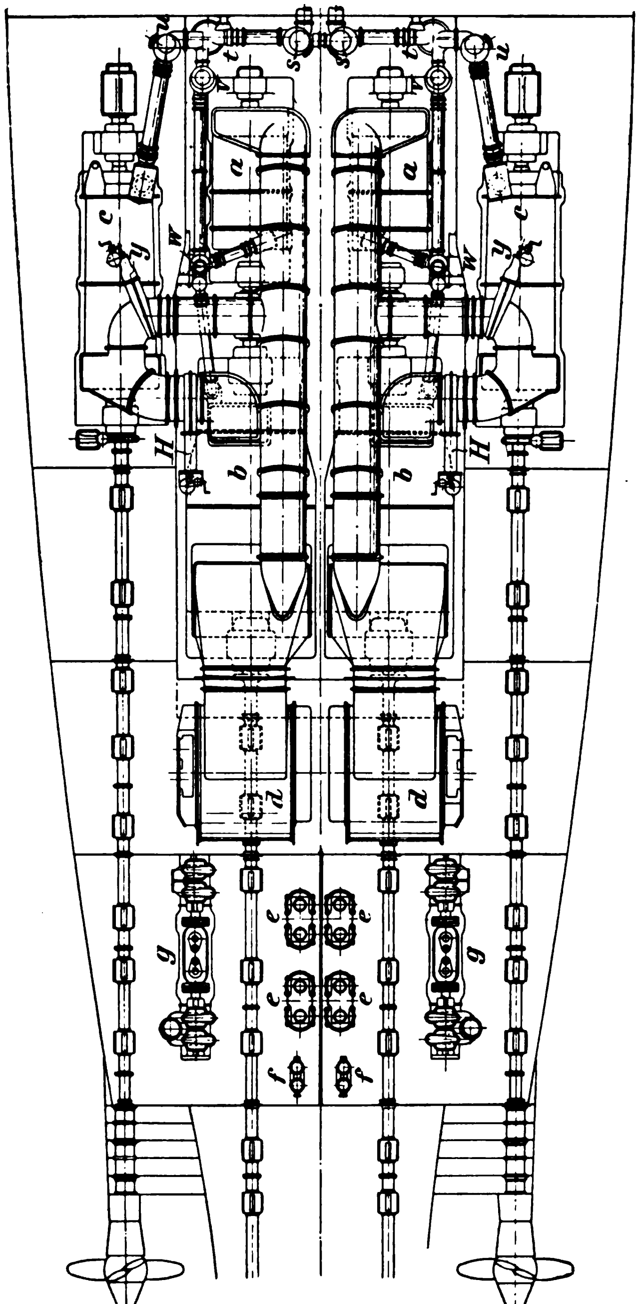


Fig. 93.

"Mauretania," based on the article in *Engineering*, 1907, pp. 636 ff. (Figs. 93 and 94).

A four shaft arrangement was chosen for this steamer. The two inner shafts are driven by the low-pressure turbine *b*, or by the reversing turbines *a*. These four turbines are placed in the main engine-room, divided from the rest of the propelling machinery by watertight bulkheads. The two wing shafts are each driven by a high-pressure turbine *c*, each turbine being placed in a separate engine-room. Aft of the main engine-room are placed the condensers *d*, one for each turbine set. The space aft of the condenser room is divided by a watertight aft and fore bulkhead into two compartments containing the four wet-air pumps *e*, the four dry-air pumps *f*, and the circulating water pump *g*. Above this compartment, on a flat at the level of the orlop deck, are the four turbo-generators, two in each of the rooms, which are divided by a fore and aft bulkhead.

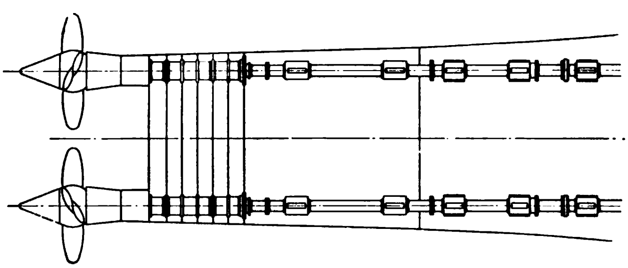


Fig. 94.

Eight feed pumps are placed against the forward engine-room bulkhead. The two wing compartments aft of the high-pressure turbines each contain an auxiliary condenser and a feed water heater. Three evaporators for boiler feed and drinking water are placed in each high-pressure turbine-room.

The boiler steam is taken by two steel pipe lines to the main valves *s* placed at the forward bulkhead. Both valves are interconnected by a short pipe length. From the valves *s* the steam passes to the separators *t*. At sea under full speed the high-pressure regulating valves *u* are open, allowing the steam to enter the high-pressure turbines *c*. From here the steam passes to the low-pressure turbines *b*, and through the exhaust steam mains to the condenser *d*.

In case of an accident to the low-pressure turbine, valve H can be closed, valve *y* opened, and the steam from the high-pressure turbine exhausted direct to the condenser without passing through the low-pressure turbine.

When entering or leaving the port where frequent manœuvring is necessary the high-pressure turbine is put out of commission by closing the regulating valves *u* and the exhaust steam valves H. The steam after passing through the separators *t* and the valves *v* is led to the manœuvring valves *w*, which enable steam to be admitted either to the low-pressure main turbine or the reversing turbines. The manœuvring valves *w* are arranged to ensure that both valves could not be opened simultaneously, but that both valves might be closed at the same time.

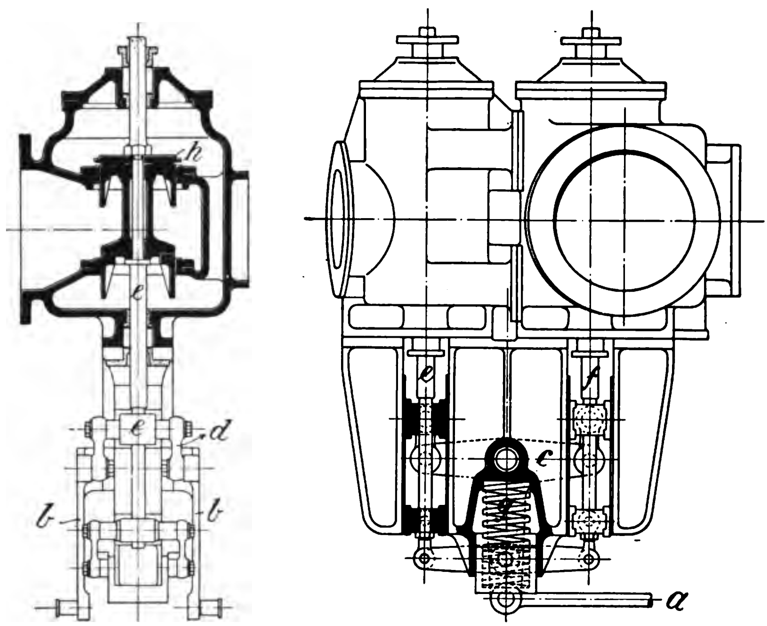


Fig. 95.

From the drawing (Fig. 95), it will be seen that on the end of the valve spindles *e* and *f* there is a lever connecting with both spindles. From the centre of the lever there is a spring *g* compressed between the valve casing and the lever which tends to keep both valves closed. Mounted on each valve spindle at a higher point is a sliding block, which is connected by links to a common lever *bc* fulcrummed in the centre. This lever *bc* is attached to the Brown steam and hydraulic engine used for actuating the gear. When the engine moves it acts through the fulcrum lever, to lift one of the sliding blocks until it comes into contact with a shoulder on one spindle, whereby one of the valves is opened. The other end of the fulcrum lever being depressed, pulls the block attached to it downwards, and as the latter rides loosely on the spindle, this valve is not affected. The return stroke of the engine brings the fulcrummed lever to the horizontal position, pulling the block down the spindle. At the same time the spring closes the valve.

The governor gear in connection with the main valves is interesting.

This is of the Aspinall type driven off the forward end of each line of main shafting through a worm and worm-wheel keyed to a shaft at right angles to the main shaft. From one end of this is taken the crank and connecting rod for driving the lever carrying the Aspinall governor, whilst from the other end there is taken the drive of the tachometers. The governor makes half the number of revolutions of the turbine. In the event of excessive speed, pawls on the governor come into contact with a trip lever, which is connected to the horizontal shaft extending across the engine-room, and actuating through a lever the valves of the Brown engine for the main steam valves.

Between the trip lever of each turbine governing gear and the shaft across the engine-room is a slotted end, so that in the event of any one turbine exceeding the speed, it would operate the shaft controlling the stop valve without disturbing the governor gear of any of the other turbines. By similar means the Brown engine can be set in motion to open the main valve without being connected to the governor gear of the turbine which has exceeded the speed limit. Both valves may be closed simultaneously in emergency, but either valve may be opened by hand.

The stop valves *s* on the engine-room side of the bulkhead, the high-pressure main stop valves *u*, and the manœuvring valve *w* are balanced valves with cast-steel bodies. A perforated brass strainer in a cast-steel casing is fitted at the steam inlet to each turbine.

The exhaust valve H from the low-pressure turbine is of the sluice type of 75 in. internal diameter, the exhaust valve from the high-pressure turbine direct to the condenser of the same type, 60 in. internal diameter. Both valves are of the same design. The valve H is operated by a 12 H.P. motor, and is also arranged for hand operation.

The valves are worked from the starting pedestal, of which there are two, one for each half of the engine-room.

Fig. 96 gives an engraving of the port pedestal. The inner wheel β works through mitre wheels, spindles and screw the valves of the Brown engine for controlling the manœuvring valve *w*. A connecting rod is driven by one of the sliding blocks, which shows on the scale *E* on the starting pedestal the position of the valve *w*. In the event of the Brown engine breaking down, there is a hand-pump γ on the starting pedestal which enables the manœuvring valves to be operated. The large hand-wheel α on the starting pedestal works the high-pressure regulating valve *u*. This valve has also an auxiliary hand gear in the high-pressure turbine-room, which operates the same spindle by a bevel gear.

The exhaust valves H and *y* are also opened from the starting platform by means of the two levers *s*, the position of the valve being indicated on ϵ. Two minutes are required to close the 75-in. valve. The three levers δ operate the drain cocks.

To allow for the expansion of the turbines and exhaust pipes, these are fitted with copper expansion pipes where they pass through

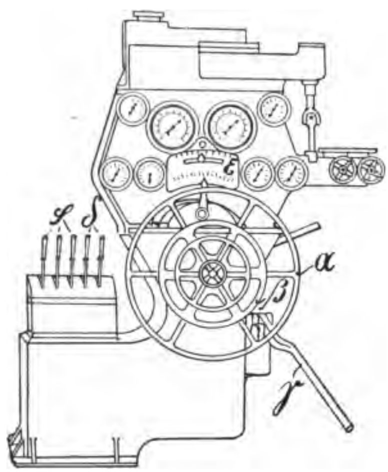


Fig. 96.

the longitudinal bulkhead, as illustrated in Fig. 97. This allows for any movement, either horizontal or vertical, due to expansion or contraction.

To enable the pressure in the various stages of the turbines to be ascertained, each turbine is fitted with two gauges fitted with a three or four way cock, so that by turning a handle the pressure at each stage of expansion can be indicated on the gauge dial.

§ 133. **Passenger Steamer with A.E.G. Turbines.**—Unfortunately, the mercantile marine has not availed itself of the advantageous arrangement which the A.E.G. turbine permits, and up to the present the only installation is that of the T.S.S. "Kaiser."

This vessel was built in 1904 by the Vulcan Company Ltd. at Stettin, and is equipped with two turbines built by the A.E.G., each of 3,300 H.P. at 560 revs. per min. Fig. 35, page 98, is a sectional view of the turbines, and Fig. 98 shows the arrangement in the vessel. The engine-room contains the two turbines, each exhausting through two riveted steel pipes of circular section to a condenser. Each condensing plant is designed to maintain a vacuum

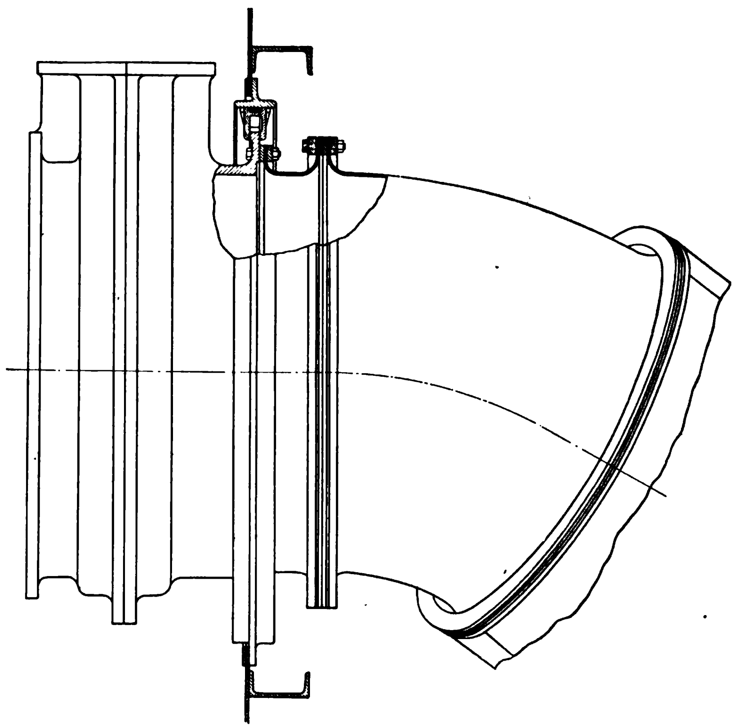


Fig. 97.

of 94 to 95 per cent. of the barometer, and includes in addition to the condenser a steam-driven circulating water pump *a*, the dry-air pump *t* driven by the same engine (see § 119, Fig. 81), and a small water pump *f* for withdrawing the condensed water from the condenser (§ 119).

For each turbine an oil pump *o* is provided, this pump being of the centrifugal type instead of a duplex pump, as usually employed in naval boats.

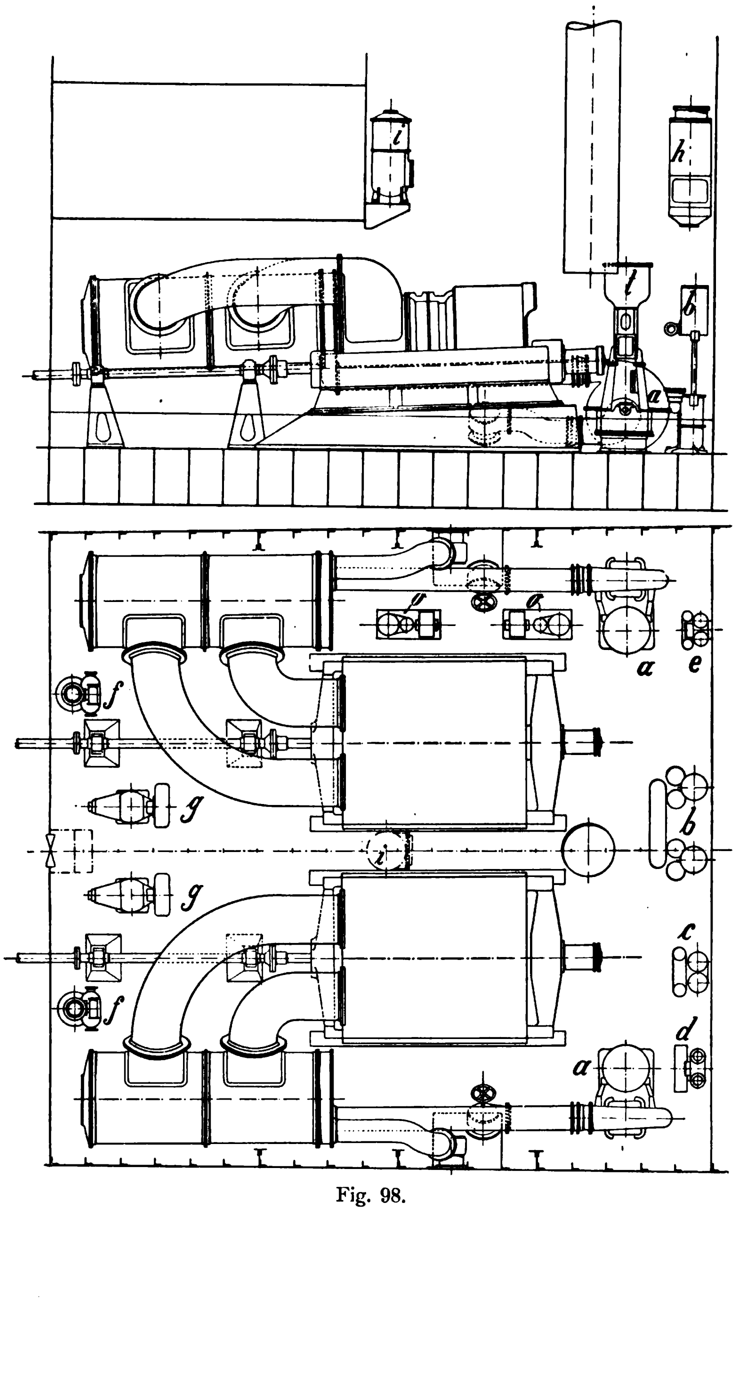


Fig. 98.

The engine-room contains :—

One set of main feed pumps - - -	*b*
One set of auxiliary feed pumps - -	*c*
—One bilge and ballast pump - - -	*d*
One sanitary pump - - - -	*e*
Two turbine-driven electric light engines -	*g*
One evaporator - - - - -	*i*
One feed heater - - - -	*h*

In addition to being a turbine-propelled vessel, it is interesting on account of being one of the few passenger steamers fitted with water-tube boilers. These are four in number, of the Yarrow-Vulcan type fitted with Howden's forced draught, the total heating surface being 1,696 sq. m. (18,250 sq. ft.), and a grate area of 32 sq. m. (345 sq. ft.).

The boilers have proved a success in every way.

PART VIII.

General Remarks on the Arrangement of Steam Turbines in Steamers.

§ 134. In arranging the machinery in a vessel the following points must be observed :—

1. It must be possible to raise the cover of the turbine high enough to permit of inspection of the interior of the turbine, and in designing the plant it is necessary to make certain that the condenser in plan view nowhere covers the turbine.
2. It must be possible to withdraw the condenser tubes without having to remove any important parts of machinery.
3. It must be possible to remove the exhaust pipe connections to give access for lifting the turbine cover.
4. The air pump must be placed so low that the condensed steam water of the low-pressure turbines flows to the air pump, otherwise separate drainage becomes necessary (see § 95).
5. Allowance must be made for the expansion and contraction of the turbine, exhaust mains, condensers, and steam mains.
6. The oil pump must be easily accessible from the starting platform, from where the water gauges of the hot-water tank, the engine-room telegraph and tachometer must be clearly visible.
7. The starting platform must be well ventilated.

PART IX.

TURBINE-DRIVEN AUXILIARIES.

§ 135. **General Remarks.**—The driving of the auxiliaries on board a vessel by steam turbines instead of by ordinary reciprocating engines presents a number of advantages:—

1. Simplicity.
2. Low cost of maintenance.
3. Minimum of supervision.
4. Supply of feed water free from oil.

The practice now adopted as a standard in land installations of using electric motors for driving the auxiliaries is not likely to be adopted in naval installations, both in view of the dependency on the generating station and the switch gear, and the difficulty of repairs in case of a breakdown, and there is little doubt but that at an early date steam turbines will be universally adopted for driving all auxiliaries.

§ 136. **Air and Condensed Steam Pumps.**—These pumps can be built either with horizontal or vertical shaft.

Both pumps are combined in a single casing, the air and water inlet being separated. Separate centrifugal pumps keyed to the same shaft deal with the air and water. The turbine wheel for driving the pumps is keyed to the same shaft. The air pump has to remove the air at a very low pressure, and compress it to about one-fiftieth of its volume at atmospheric pressure. A compression of this magnitude cannot be accomplished with an air compressor of the ordinary centrifugal type, as a compressor for such a duty would have to consist of a series of impellers the number of which would be prohibitive.

The centrifugal air pump patented and employed by the A.E.G., and shown in Fig. 99, consists of an impeller running at a high speed, taking water from a tank, and discharging it into a diffuser surrounding the impeller. The space between the impeller and the diffuser is connected to the vacuum space of the condenser. The streams of water discharged from the impeller imprison the air, the mixture of air and water being discharged into a diffuser. In the diffuser the air is gradually compressed by the moving body of water, until on leaving

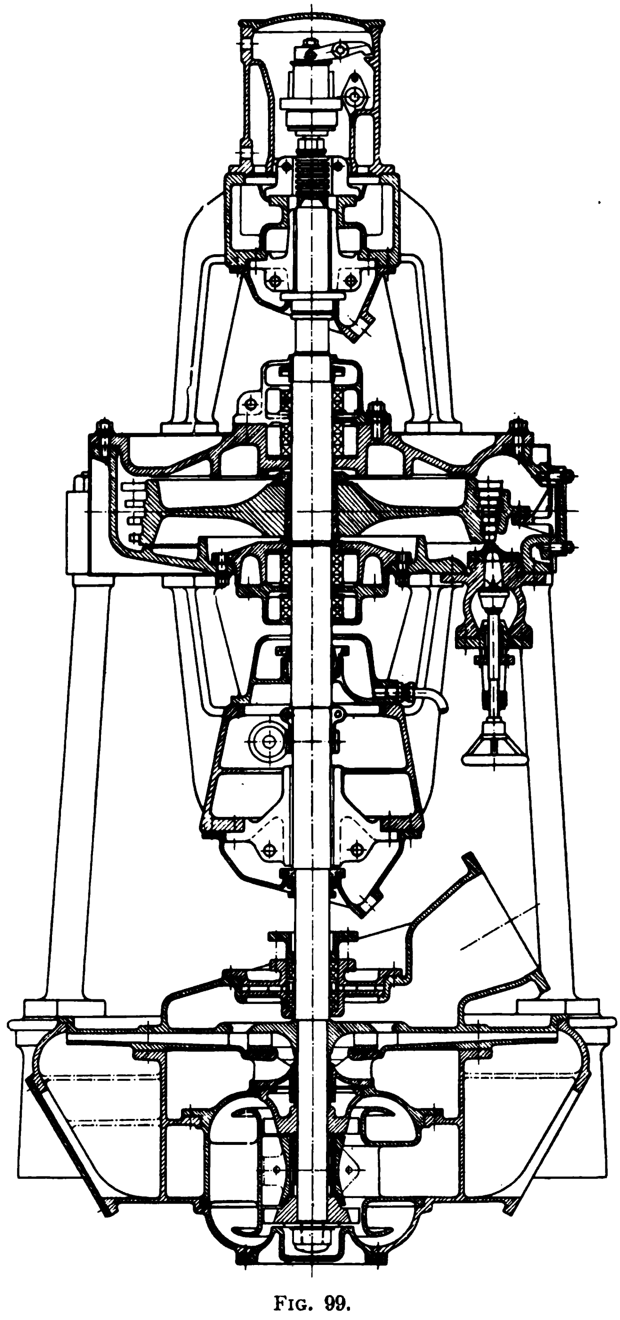


FIG. 99.

the diffuser it is compressed to atmospheric pressure. The amount of water is very much less than that required by the known type of air ejector operated by a water jet only. As the diffuser surrounds the whole impeller the entire circumference of the impeller can be utilised, entailing a small diameter, and consequently a speed sufficiently high for direct coupling to a turbine. The water after being ejected is collected in a sump and used again for the same purpose, so that the water is in continual circulation.

The water for the air ejector must not, however, be salt water, but of a nature suitable for boiler feed, as the air withdrawn from the condenser contains a certain amount of vapour (amounting to about 3 per cent. of the feed), which is condensed on coming in contact with the ejector water. A part of the ejector water is used for boiler feed. To avoid the condensed vapour heating the ejector water a surface cooler is built into the tank.

Fig. 99 shows the arrangement of a pump of this description designed to deal with 100 tons of condensed steam water per hour, but capable of dealing with twice this quantity for short periods.

The speed of the turbine is kept constant by a small governor, and a separate emergency governor is fitted to come into action if the speed exceeds a certain predetermined amount. Automatic lubrication is fitted so that the whole combination requires no attention when once set to work.

Weight is reduced wherever possible without sacrificing the requisite mechanical strength.

The tank or sump into which the pump discharges is supplied with water from the feed tanks, and a connection from the sump to the condenser is provided to return the surplus water due to the condensation of the vapour to the feed tanks.

§ 137. **Turbine-Driven Ventilating Fans.**—The high speed required for the ventilating and forced draught fan make turbine drive eminently suitable, both on account of the simplicity of the mechanism and absence of wear on the parts rotating at high speed.

Fig. 100 shows a type of turbine-driven ventilating fan for the stokehold of a destroyer. The fan wheel is keyed to a vertical shaft above the turbine, and consists of a wheel with axial air inlet discharging radially through a diffuser. The turbine is a single Curtis wheel with three rows of blades enclosed in a light casing, and keyed to the fan shaft immediately below the fan. Only two bearings are used, one above and one below the turbine wheel. Carbon packing is used for the turbine glands. The speed of this fan is comparatively low, being only 1,100 to 1,300 revs. per min., but even at this speed the

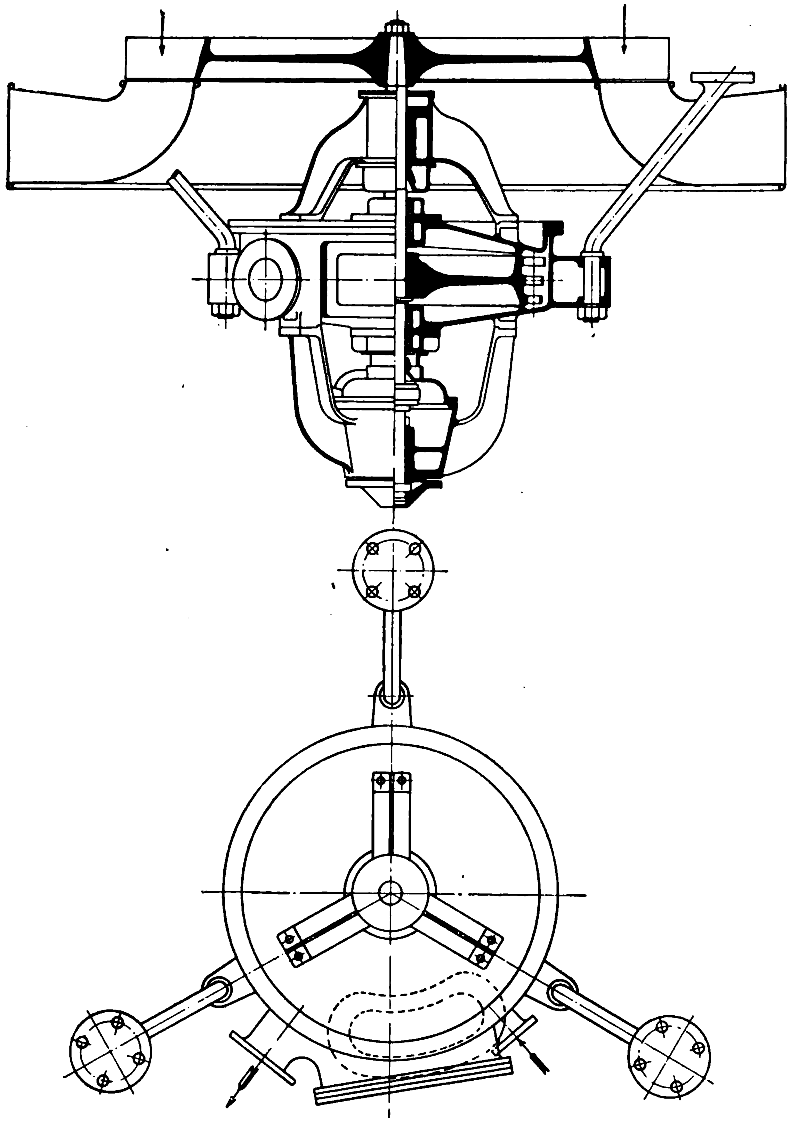


FIG. 100.

turbine has proved to be more economical than a high-speed reciprocating engine. This and similar designs which were got out by the A.E.G. have been frequently installed on board ship.

§ 138. **Turbine-Driven Circulating Pumps.**—The high speed necessary for steam turbines has led to the substitution of the ordinary centrifugal pump by a screw pump. Such a pump is shown in Fig. 101. A comparison of this pump with a centrifugal pump shows the following advantages of the former:—

1. The quantity of water discharged by the pump is less dependent on the head.
2. The efficiency is high over a greatly varying head, and the maximum efficiency is equal to that of a centrifugal pump, whereas the latter has a high efficiency over a small range of head only.

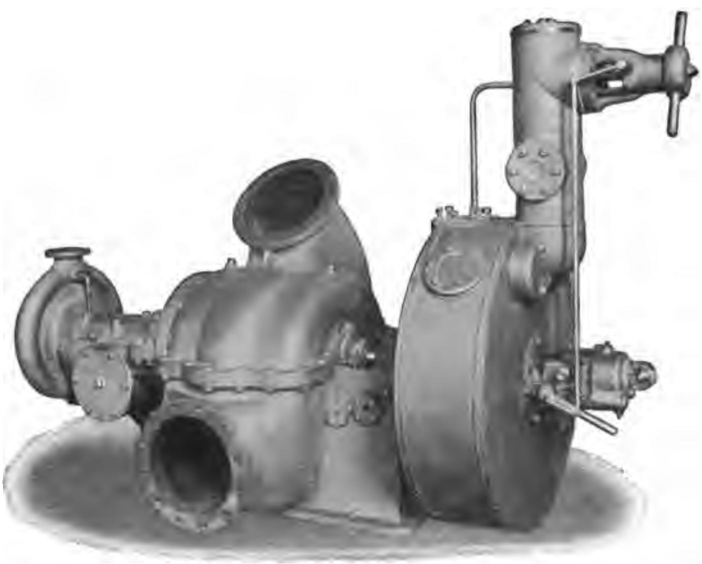


FIG. 101.

3. The power for driving the pump decreases with decreasing head, entailing no throttling of the discharge as with a centrifugal pump to maintain constant load on the motor. This has the advantage that when designing the pump no exact calculation of the head is required, but the pump can be designed for dealing with an ample quantity of water, and should on completion the head prove less than anticipated, less power will be required for driving the pump.
4. Enhanced safety as the clearance between the rotating parts and casing are less than in a centrifugal pump.

The overall dimensions of both pumps are approximately the same, as these depend in large pumps less on the speed than on the quantity of water to be dealt with. No doubt centrifugal pumps for certain duties are particularly suitable for driving by steam turbines, as this type of pump when correctly designed can be run at high speeds; but this depends in a large measure on the quantity of water to be dealt with and the pumping head.

§ 139. **Boiler Feed Pumps.**—It is to be anticipated that the present slow-speed reciprocating boiler feed pumps will gradually be replaced by high-speed turbine-driven pumps, and that this will result in a saving in space and weight. Fig. 102 shows a boiler feed pump

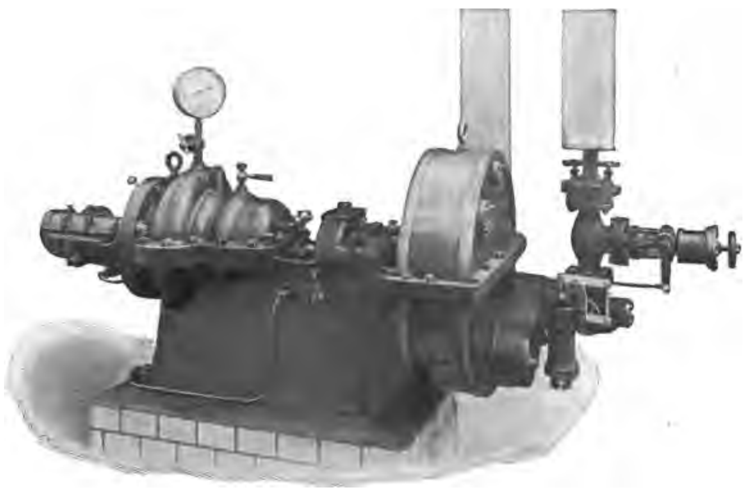


FIG. 102.

of this type. The horizontal arrangement of the set was chosen on account of accessibility, and because in this case there was no special reason for economising space by adopting a vertical arrangement. The ground space occupied by the turbo set, as well as the weight, is about the same as that of an ordinary duplex boiler feed pump. An emergency governor is fitted to the turbine. If all the feed check valves are closed, the discharge pressure will rise, and to prevent this increasing above a safe maximum, a small safety valve is fitted, allowing a sufficient quantity of water to overflow, and thus prevent the heating and evaporation of the water in the pump. A Venturi tube fitted with a mercury column is used for measuring the flow of water through the pipe line, the drop in pressure between the full area of the pipe and the restricted area at the throat of the nozzle being a measure of the quantity flowing through the pipe. (Fig. 103.)

§ 140. **Notes on Turbines for Driving the Auxiliaries.**--These comparatively small turbines are not as a rule designed to exhaust to a condenser on account of the difficulty in preventing air leakage at the glands, and also the exhaust pipe line would have to be of considerable size to reduce the drop in pressure. For these reasons the turbines are designed to exhaust against a back pressure of a few pounds to a feed

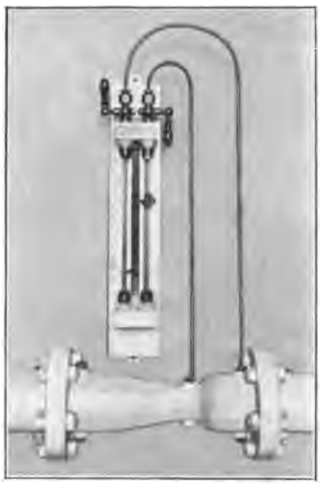


FIG. 103.

water heater. The turbines are not designed with a drum, but consist of a single Curtis wheel with two or three rows of blades (see Fig. 100). Compared to the main turbines the consumption is, of course, high, varying according to the size from 15 to 20 kg. per B.H.P., or 33·1 to 44·2 lbs. per B.H.P. This, however, is still considerably less than the consumption of auxiliaries driven by reciprocating engines, especially of direct-acting boiler feed pumps, for which the steam consumption is frequently as much as 30 to 50 kg. per B.H.P., or 66·2 to 110 lbs. per B.H.P. and more.

PART X.

STEAM TABLES.

TABLE 16.—*Properties of Saturated Steam (Metric).*

1 Pressure (kg. per sq. cm.) p	2 Temperature. t	3 Volume of 1 kg. Steam. cb. m. v''	4 Weight of 1 cb. m. Steam. kg. $\frac{1}{v''}=\gamma''$	5 Heat of the Liquid. Cal. h	6 Heat Absorbed in Overcoming *Internal* Resistance to Vaporisation. Cal. ρ	7 Heat Absorbed in Overcoming *External* Resistance to Vaporisation. Cal. ψ	8 Total Heat of Steam, 5+6+7. Cal. i''
0·02	17·3	68·126	0·01468	17·3	553·6	31·91	602·9
0·04	28·8	35·387	0·02826	28·8	546·3	33·15	608·3
0·06	36·0	24·140	0·04142	36·0	541·7	33·92	611·6
0·08	41·3	18·408	0·05432	41·4	538·2	34·49	614·1
0·10	45·6	14·920	0·06703	45·7	535·4	34·94	616·0
0·12	49·2	12·568	0·07956	49·3	533·1	35·32	617·7
0·15	53·7	10·190	0·09814	53·8	530·1	35·79	619·7
0·20	59·8	7·777	0·12858	59·9	526·1	36·42	622·4
0·25	64·6	6·307	0·1586	64·8	522·9	36·92	624·6
0·30	68·7	5·316	0·1881	68·9	520·2	37·34	626·4
0·35	72·3	4·600	0·2174	72·5	517·8	37·70	628·0
0·40	75·5	4·060	0·2463	75·7	515·6	38·02	629·4
0·50	80·9	3·2940	0·3036	81·2	512·0	38·56	631·7
0·60	85·5	2·7770	0·3601	85·8	508·8	39·01	633·7
0·70	89·5	2·4040	0·4160	89·9	506·1	39·39	635·3
0·80	93·0	2·1216	0·4713	93·5	503·6	39·73	636·8
0·90	96·2	1·9003	0·5262	96·7	501·4	40·03	638·1
1·0	99·1	1·7220	0·5807	99·6	499·4	40·30	639·3
1·1	101·8	1·5751	0·6349	102·3	497·5	40·55	640·7
1·2	104·2	1·4521	0·6887	104·8	495·7	40·78	641·3
1·4	108·7	1·2571	0·7955	109·4	492·6	41·18	643·1
1·6	112·7	1·1096	0·9013	113·4	489·7	41·54	644·7
1·8	116·3	0·9939	1·0062	117·1	487·1	41·85	646·0

TABLE 16.—*Properties of Saturated Steam (Metric)*—*continued.*

1 Pressure (kg. per sq. cm.). p	2 Temperature. t	3 Volume of 1 kg. Steam. cb. m. v''	4 Weight of 1 cb. m. Steam. kg. $\frac{1}{v''}=\gamma''$	5 Heat of the Liquid. Cal. h	6 Heat Absorbed in Overcoming *Internal* Resistance to Vaporisation. Cal. ρ	7 Heat Absorbed in Overcoming *External* Resistance to Vaporisation. Cal. ψ	8 Total Heat of Steam, 5+6+7. Cal. i''
2·0	119·6	0·9006	1·1104	120·4	484·7	42·14	647·2
2·5	126·7	0·7310	1·3680	127·7	479·4	42·74	649·9
3·0	132·8	0·6163	1·6224	133·9	474·9	43·23	652·0
3·5	138·1	0·5335	1·8743	139·4	470·8	43·65	653·8
4·0	142·8	0·4708	2·1239	144·2	467·2	44·01	655·4
4·5	147·1	0·4217	2·3716	148·6	463·9	44·33	656·8
5·0	151·0	0·3820	2·6177	152·6	460·8	44·61	658·1
5·5	154·6	0·3494	2·8624	156·3	458·0	44·87	659·2
6·0	157·9	0·3220	3·1058	159·8	455·3	45·10	660·2
6·5	161·1	0·2987	3·3481	163·0	452·8	45·32	661·1
7·0	164·0	0·2786	3·5891	166·1	450·4	45·51	662·0
7·5	166·8	0·2611	3·8294	168·9	448·2	45·67	662·8
8·0	169·5	0·2458	4·0683	171·7	446·0	45·86	663·5
8·5	172·0	0·2322	4·3072	174·3	443·9	46·02	664·2
9·0	174·4	0·2200	4·5448	176·8	441·9	46·17	664·9
9·5	176·7	0·2091	4·7819	179·2	440·0	46·30	665·5
10·0	178·9	0·1993	5·018	181·5	438·2	46·43	666·1
11·0	183·1	0·1822	5·489	185·8	434·6	46·67	667·1
12·0	186·9	0·1678	5·960	189·9	431·3	46·88	668·1
13·0	190·6	0·15565	6·425	193·7	428·2	47·08	668·9
14·0	194·0	0·14515	6·889	197·3	425·2	47·26	669·7
15·0	197·2	0·13601	7·352	200·7	422·4	47·43	670·5
16·0	200·3	0·12797	7·814	203·9	419·7	47·58	671·2
18·0	206·1	0·11450	8·734	210·0	414·6	47·85	672·4
20·0	211·3	0·10365	9·648	215·5	409·8	48·08	673·4

TABLE 17.—*Properties of Saturated Steam* (*English*).

Press. Lbs. p	Temperature. Deg. Fahr. t	Pressure. Atmos.	Specific Volume. Cub. ft. per lb. v	Density. Lbs. per cub. ft. $\frac{1}{v}$	Heat of the Liquid. h	Latent Heat of Evaporation. L	Total Heat of Steam. H
1	101·83	0·068	333·0	0·00300	69·8	1034·6	1104·4
2	126·15	0·136	173·5	0·00576	94·0	1021·0	1115·0
3	141·52	0·204	118·5	0·00845	109·4	1012·3	1121·6
4	153·01	0·272	90·5	0·01107	120·9	1005·7	1126·5
5	162·28	0·340	73·33	0·01364	130·1	1000·3	1130·5
6	170·06	0·408	61·89	0·01616	137·9	995·8	1133·7
7	176·85	0·476	53·56	0·01867	144·7	991·8	1136·5
8	182·86	0·544	47·27	0·02115	150·8	988·2	1139·0
9	188·27	0·612	42·36	0·02361	156·2	985·0	1141·1
10	193·22	0·680	38·38	0·02606	161·1	982·0	1143·1
11	197·75	0·748	35·10	0·02849	165·7	979·2	1144·9
12	201·96	0·816	32·36	0·03090	169·9	976·6	1146·5
13	205·87	0·885	30·03	0·03330	173·8	974·2	1148·0
14	209·55	0·953	28·02	0·03569	177·5	971·9	1149·4
15	213·0	1·021	26·27	0·03806	181·0	969·7	1150·7
20	228·0	1·361	20·08	0·04980	196·1	960·0	1156·2
25	240·1	1·701	16·30	0·0614	208·4	952·0	1160·4
30	250·3	2·041	13·74	0·0728	218·8	945·1	1163·9
35	259·3	2·382	11·89	0·0841	227·9	938·9	1166·8
40	267·3	2·722	10·49	0·0953	236·1	933·3	1169·4
45	274·5	3·062	9·39	0·1065	243·4	928·2	1171·6
50	281·0	3·402	8·51	0·1175	250·1	923·5	1173·6
55	287·1	3·742	7·78	0·1285	256·3	919·0	1175·4
60	292·7	4·083	7·17	0·1394	262·1	914·9	1177·0
65	298·0	4·423	6·65	0·1503	267·5	911·0	1178·5
70	302·9	4·763	6·20	0·1612	272·6	907·2	1179·8
75	307·6	5·103	5·81	0·1721	277·4	903·7	1181·1
80	312·0	5·444	5·47	0·1829	282·0	900·3	1182·3
85	316·3	5·784	5·16	0·1937	286·3	897·1	1183·4
90	320·3	6·124	4·89	0·2044	290·5	893·9	1184·4
95	324·1	6·464	4·65	0·2151	294·5	890·9	1185·4
100	327·8	6·80	4·429	0·2258	298·3	888·0	1186·3
105	331·4	7·14	4·230	0·2365	302·0	885·2	1187·2
110	334·8	7·49	4·047	0·2472	305·5	882·5	1188·0
115	338·1	7·83	3·880	0·2577	309·0	879·8	1188·8
120	341·3	8·17	3·726	0·2683	312·3	877·2	1189·6
125	344·4	8·50	3·583	0·2791	315·5	874·7	1190·3
130	347·4	8·85	3·452	0·2897	318·6	872·3	1191·0
135	350·3	9·19	3·331	0·3002	321·7	869·9	1191·6
140	353·1	9·53	3·219	0·3107	324·6	867·6	1192·2
145	355·8	9·87	3·112	0·3213	327·4	865·4	1192·8
150	358·5	10·21	3·012	0·3320	330·2	863·2	1193·4
155	361·0	10·55	2·920	0·3425	332·9	861·0	1194·0
160	363·6	10·89	2·834	0·3529	335·6	858·8	1194·5
165	366·0	11·23	2·753	0·3633	338·2	856·8	1195·0
170	368·5	11·57	2·675	0·3738	340·7	854·7	1195·4
175	370·8	11·91	2·602	0·3843	343·2	852·7	1195·9
180	373·1	12·25	2·533	0·3948	345·6	850·8	1196·4
185	375·4	12·59	2·468	0·4052	348·0	848·8	1196·8

TABLE 17.—*Properties of Saturated Steam—continued.*

Press. Lbs. p	Temperature. Deg. Fahr. t	Pressure. Atmos.	Specific Volume. Cub. ft. per lb. v	Density. Lbs. per cub. ft. $\frac{1}{v}$	Heat of the Liquid. h	Latent Heat of Evaporation. L	Total Heat of Steam. H
190	377·6	12·93	2·406	0·4157	350·4	846·9	1197·3
195	379·8	13·27	2·346	0·4262	352·7	845·0	1197·7
200	381·9	13·61	2·290	0·437	354·9	843·2	1198·1
205	384·0	13·95	2·237	0·447	357·1	841·4	1198·5
210	386·0	14·29	2·187	0·457	359·2	839·6	1198·8
215	388·0	14·63	2·138	0·468	361·4	837·9	1199·2
220	389·9	14·97	2·091	0·478	363·4	836·2	1199·6
225	391·9	15·31	2·046	0·489	365·5	834·4	1199·9
230	393·8	15·65	2·004	0·499	367·5	832·8	1200·2
235	395·6	15·99	1·964	0·509	369·4	831·1	1200·6
240	397·4	16·33	1·924	0·520	371·4	829·5	1200·9
245	399·3	16·67	1·887	0·530	373·3	827·9	1201·2
250	401·1	17·01	1·850	0·541	375·2	826·3	1201·5
260	404·5	17·69	1·782	0·561	378·9	823·1	1202·1
270	407·9	18·37	1·718	0·582	382·5	820·1	1202·6
280	411·2	19·05	1·658	0·603	386·0	817·1	1203·1
290	414·4	19·73	1·602	0·624	389·4	814·2	1203·6
300	417·5	20·41	1·551	0·645	392·7	811·3	1204·1

From "Tables and Diagrams," by Lionel S. Marks and Harvey N. Davis, published by Longmans, Green, & Co. (1909).

Table 18.

Values for the Calculation of the Heat of Evaporation, &c., of Saturated Steam.

t	Y	X	Z	t	Y	X	Z
25	0·056	6	0·015	200	0·0120	1·20	0·00198
30	0·053	5	0·014	205	0·0116	1·15	0·00189
35	0·050	5	0·013	210	0·0112	1·11	0·00181
40	0·048	5	0·012	215	0·0108	1·08	0·00173
45	0·045	5	0·011	220	0·0105	1·04	0·00166
50	0·043	4·3	0·0103	225	0·0101	1·00	0·00159
55	0·041	4·1	0·0097	230	0·0098	0·97	0·00152
60	0·039	3·9	0·0091	235	0·0095	0·94	0·00146
65	0·037	3·7	0·0085	240	0·0092	0·91	0·00139
70	0·035	3·5	0·0080	245	0·0089	0·88	0·00134
75	0·033	3·4	0·0075	250	0·0086	0·85	0·00128
80	0·032	3·2	0·0070	255	0·0083	0·82	0·00123
85	0·030	3·1	0·0066	260	0·0081	0·79	0·00118
90	0·029	2·9	0·0062	265	0·0078	0·77	0·00113
95	0·028	2·8	0·0059	270	0·0076	0·75	0·00109
100	0·0265	2·66	0·00554	275	0·0074	0·72	0·00105
105	0·0255	2·55	0·00524	280	0·0071	0·70	0·00101
110	0·0243	2·44	0·00494	285	0·0069	0·68	0·00097
115	0·0232	2·35	0·00468	290	0·0067	0·66	0·00093
120	0·0223	2·24	0·00442	295	0·0065	0·64	0·00090
125	0·0214	2·14	0·00418	300	0·0063	0·62	0·00086
130	0·0205	2·06	0·00397	305	0·0062	0·60	0·00083
135	0·0197	1·97	0·00376	310	0·0060	0·58	0·00080
140	0·0189	1·89	0·00357	315	0·0058	0·57	0·00077
145	0·0181	1·82	0·00339	320	0·0057	0·55	0·00074
150	0·0174	1·75	0·00321	325	0·0055	0·53	0·00072
155	0·0168	1·68	0·00306	330	0·0053	0·52	0·00069
160	0·0161	1·61	0·00291	335	0·0052	0·50	0·00067
165	0·0155	1·55	0·00277	340	0·0051	0·49	0·00064
170	0·0149	1·49	0·00263	345	0·0049	0·48	0·00062
175	0·0144	1·44	0·00251	350	0·0048	0·46	0·00060
180	0·0139	1·38	0·00239	355	0·0047	0·45	0·00058
185	0·0134	1·33	0·00228	360	0·0046	0·44	0·00056
190	0·0129	1·29	0·00217	365	0·0044	0·43	0·00054
195	0·0124	1·24	0·00208	370	0·0043	0·41	0·00052

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The references are to pages.

Printed at THE DARIEN PRESS, *Edinburgh.*